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[54] **SPINDLE MOTOR WITH HYBRID AIR/OIL HYDRODYNAMIC BEARING**

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[73] Assignee: **International Business Machines Corporation**, Armonk, N.Y.

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[51] **Int. Cl.**[7] **F16C 32/06**

[52] **U.S. Cl.** **384/107; 384/100**

[58] **Field of Search** 384/100, 107, 384/112

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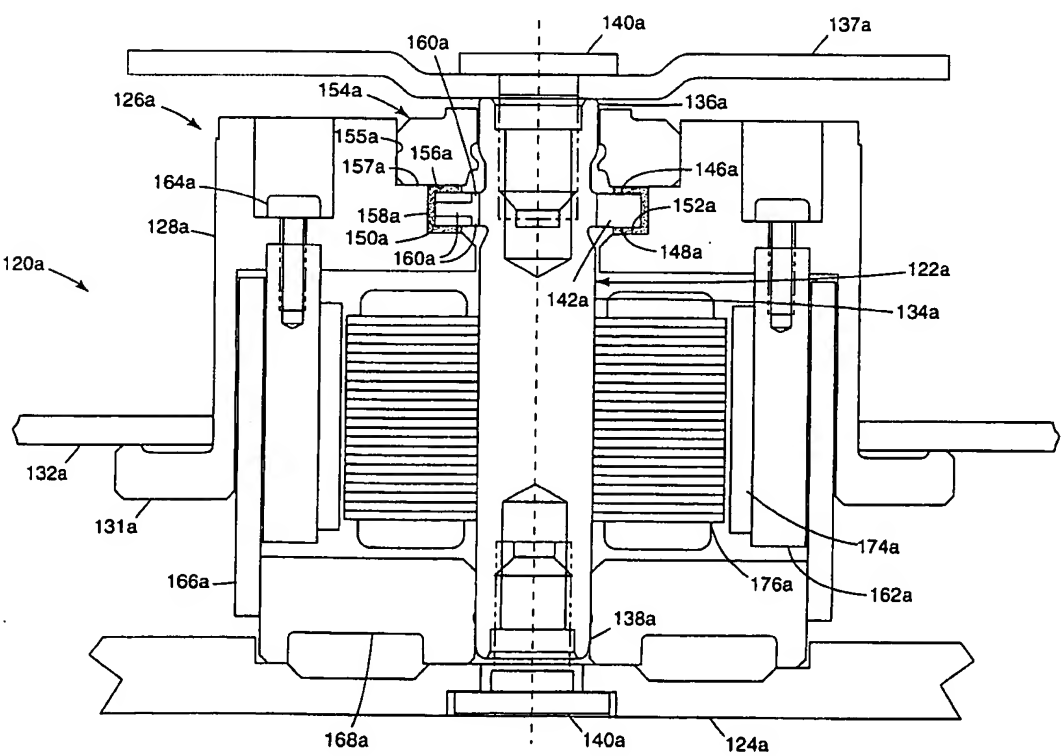
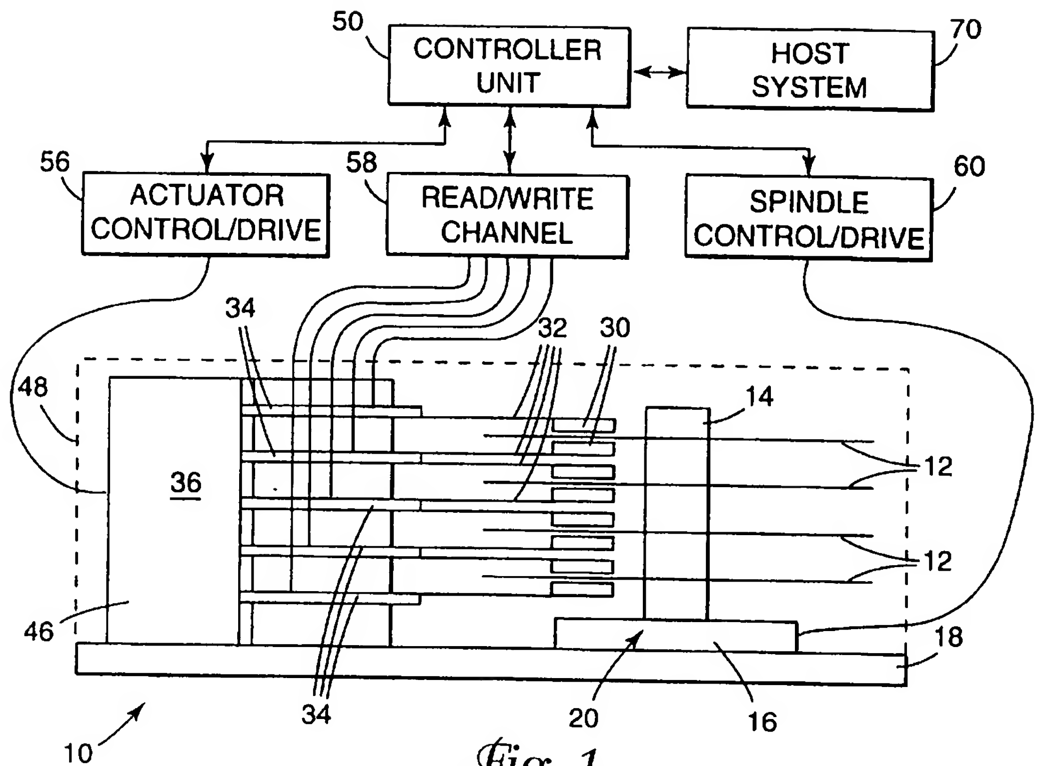
[57] **ABSTRACT**

The present disclosure relates to a spindle motor including a shaft having an outer circumferential surface. The spindle motor also includes a thrust plate fixedly connected to the shaft. The thrust plate projects radially outward from the outer circumferential surface of the shaft and includes top and bottom surfaces. The spindle motor also includes a hub assembly mounted on the shaft. The hub assembly is adapted for mounting a storage disk. The spindle motor further includes a liquid hydrodynamic bearing and an aerodynamic bearing. The liquid hydrodynamic bearing is formed along the top and bottom surfaces of the thrust plate and is adapted for transferring loads in an axial direction relative to the shaft. The aerodynamic bearing is formed along a portion of the hub assembly and is adapted for transferring loads in a radial direction relative to the shaft.

21 Claims, 8 Drawing Sheets

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HOST SYSTEM
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ACTUATOR CONTROL/DRIVE
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READ/WRITE CHANNEL
SPINDLE CONTROL/DRIVE
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Fig. 1

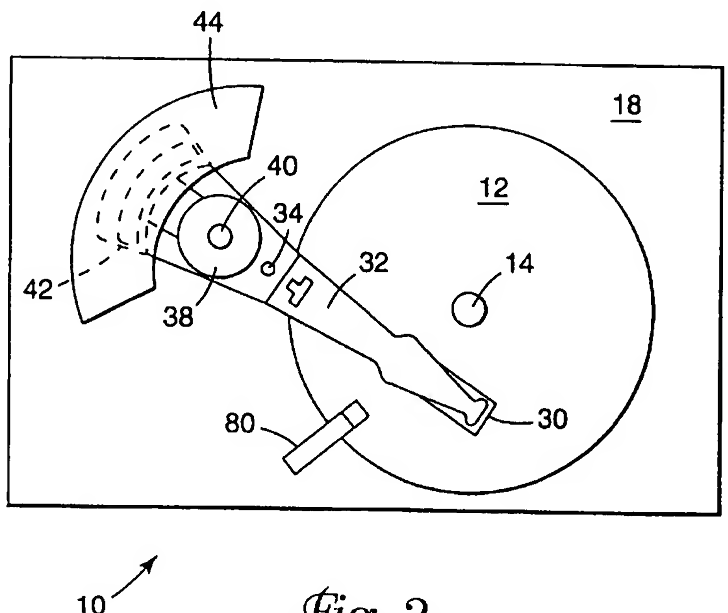
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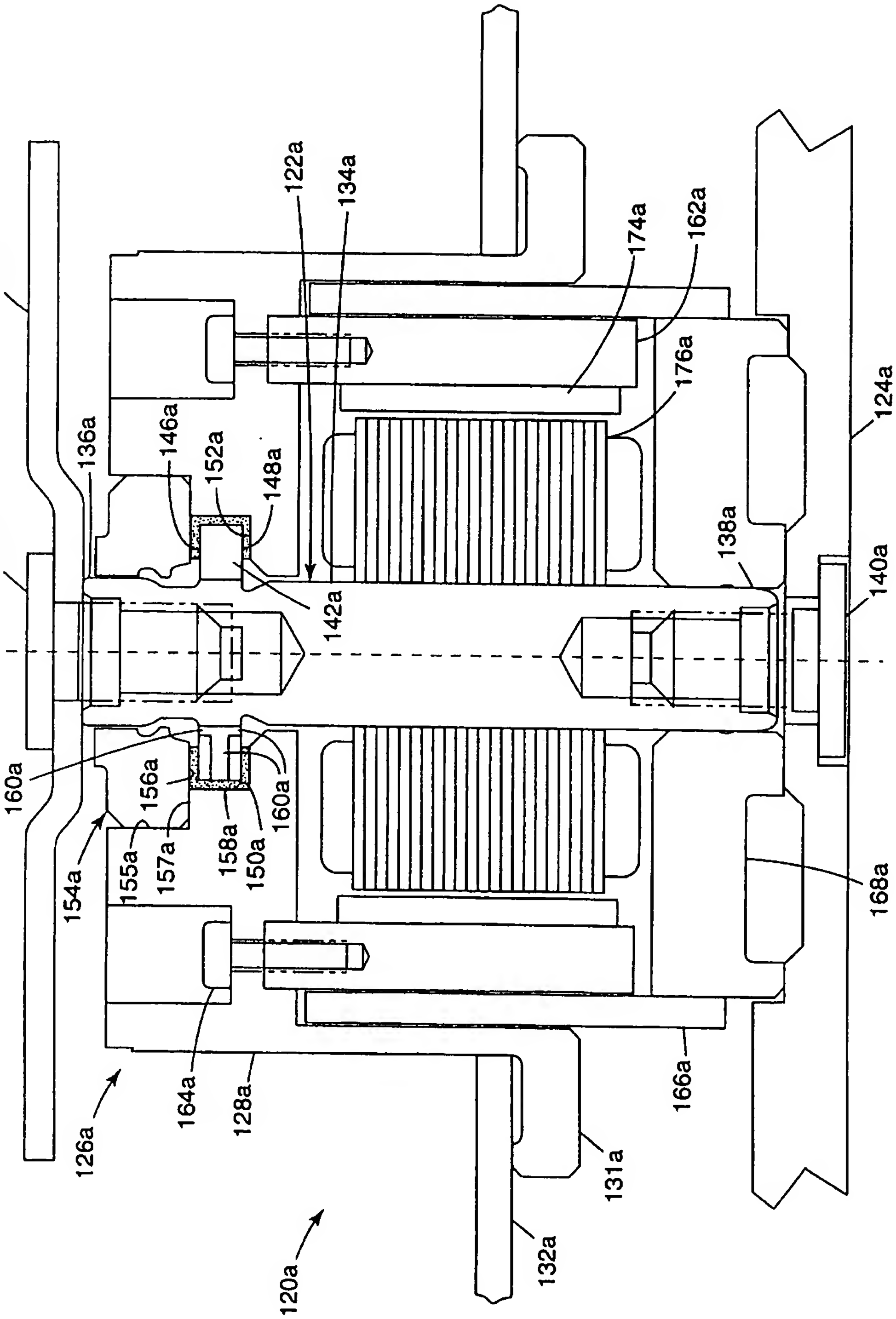
Fig. 2



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146a
152a
148a
122a
134a
174a
162a
176a
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138a
140a
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160a
156a
160a
155a
157a
158a
150a
154a
168a
164a
128a
166a
126a
131a
132a
120a

Fig. 3

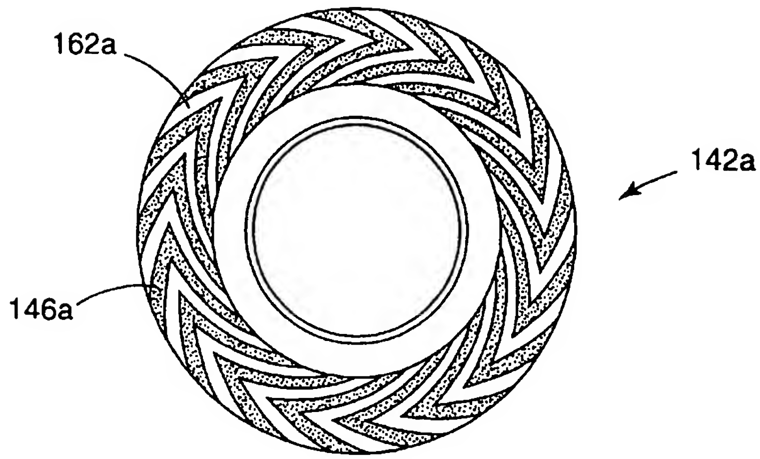


Fig. 4

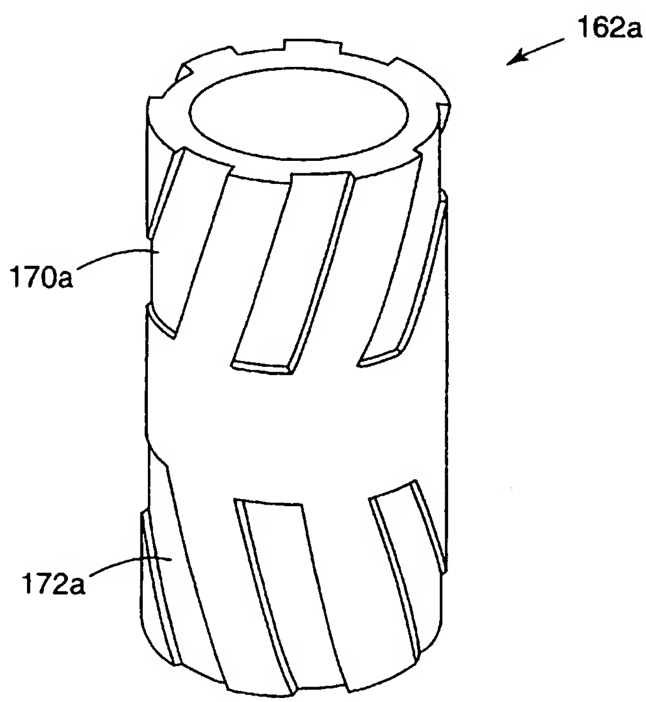


Fig. 5

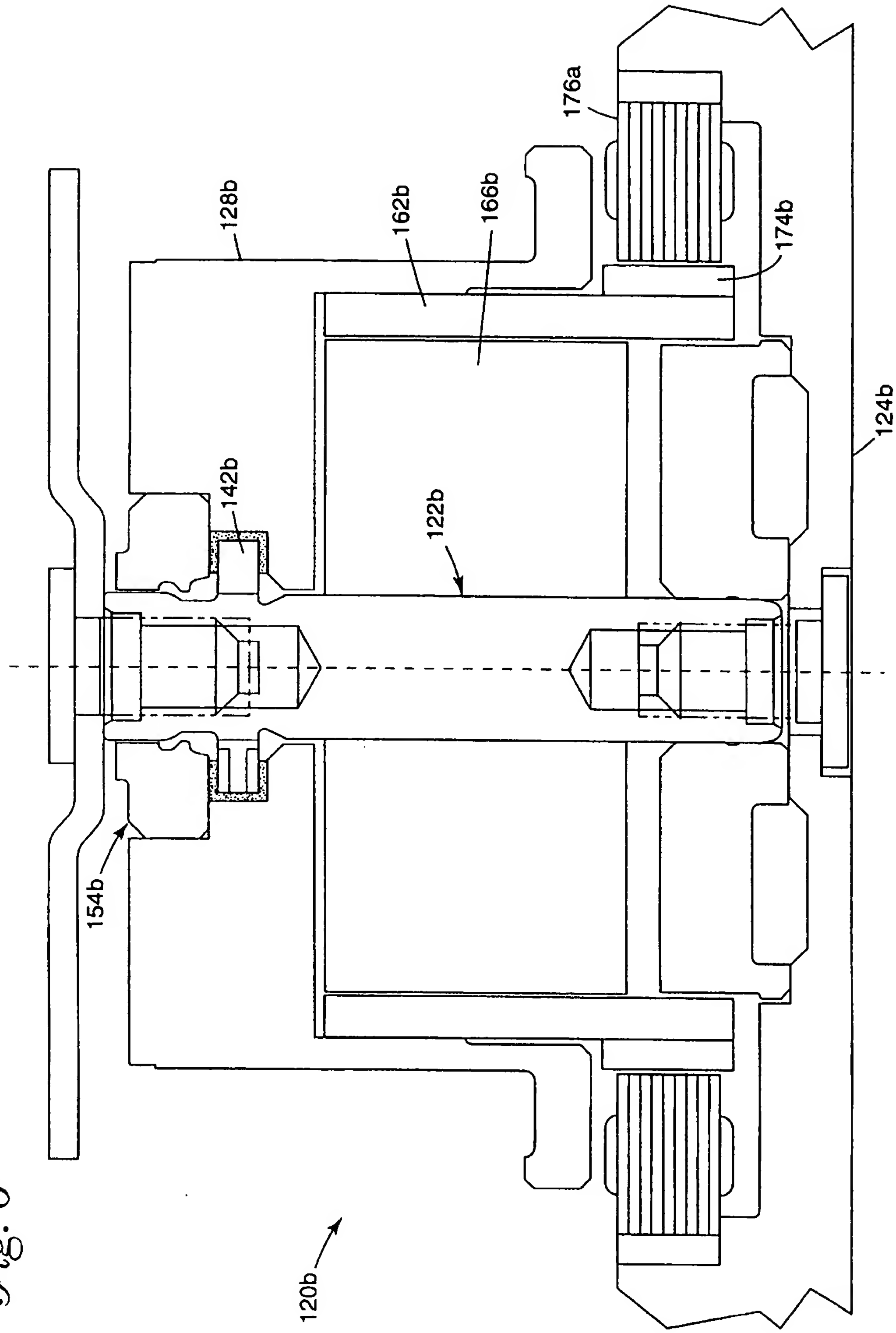


Fig. 6

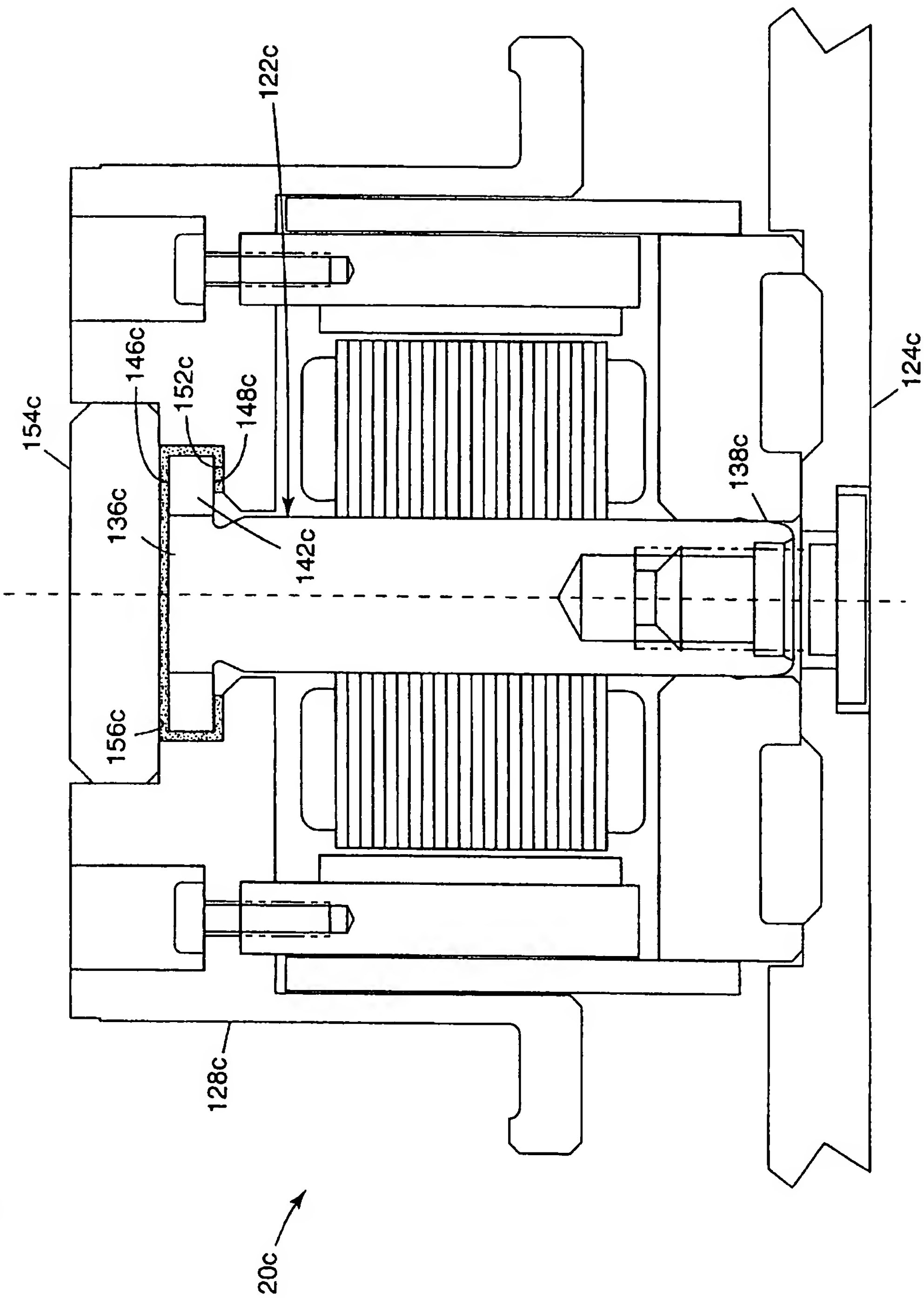


Fig. 7

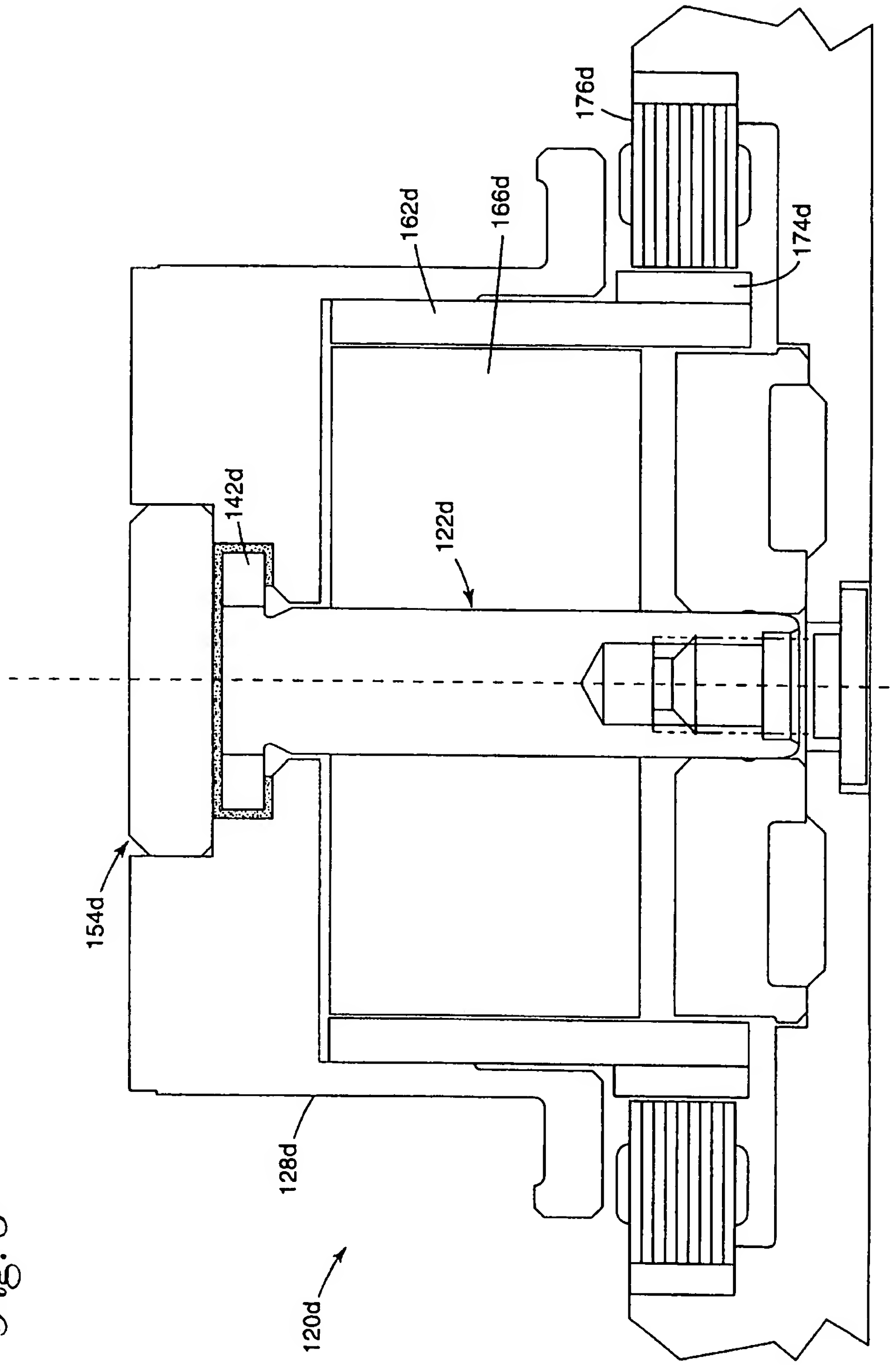
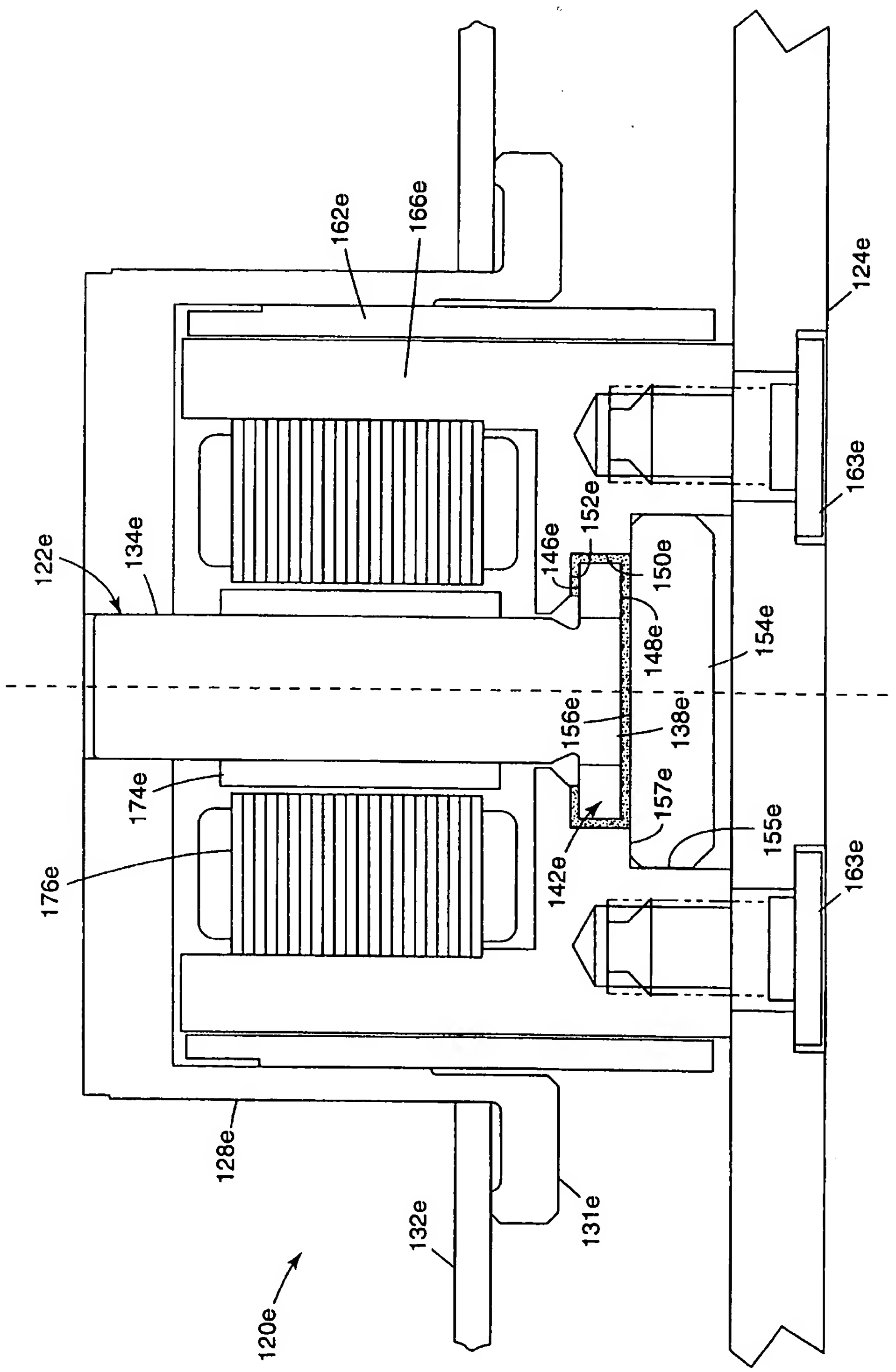


Fig. 8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132e
131e
120e

Fig. 9

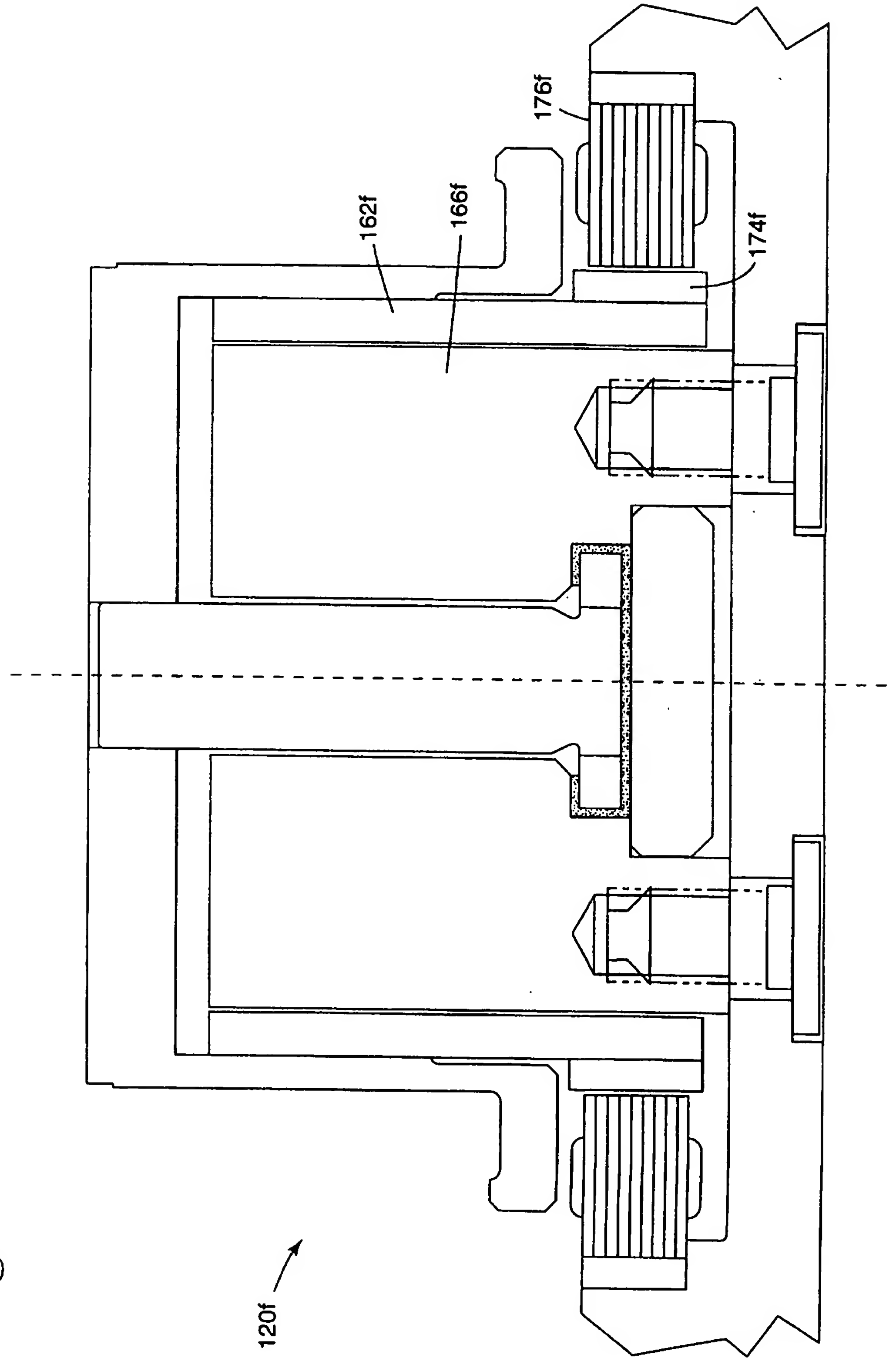

Fig. 10

SPINDLE MOTOR WITH HYBRID AIR/OIL HYDRODYNAMIC BEARING

BACKGROUND OF THE INVENTION

1. Field of the Invention

This invention relates in general to spindle motors for use in magnetic disc storage systems. More particularly, this invention relates to magnetic disc storage systems having spindle motors that use hydrodynamic bearings.

2. Description of Related Art

Data storage systems, such as disk drives, commonly make use of rotating storage disks. The storage disks are commonly magnetic disks but could also be optical. In a typical magnetic disk drive, a magnetic disk rotates at high speed and a transducing head uses air pressure to "fly" over the top surface of the disk. The transducing head records information on the disk surface by impressing a magnetic field on the disk. Information is read back using the head by detecting magnetization of the disk surface. The magnetic disk surface is divided in a plurality of concentric tracks. By moving the transducing head radially across the surface of the disk, the transducing head can read information from or write information to different tracks of the magnetic disk.

Spindle motors are commonly used to rotate magnetic disks at high speeds. Frequently, conventional spindle motors comprise small electric motors equipped with standard ball bearings. However, electric motors having ball bearings are known to experience problems such as runout or vibration that can prevent information from being accessed from disks rotated by the motors. This is especially true as advancements in data storage technology have increased magnetic disk storage densities. To overcome the problems associated with ball bearing electric motors, some disk drive systems now make use of electric motors having fluid hydrodynamic bearings. Bearings of this type are shown in U.S. Pat. No. 5,427,546 to Hensel, U.S. Pat. No. 5,516,212 to Titcomb and U.S. Pat. No. 5,707,154 to Ichiyama.

An exemplary hydrodynamic bearing typically includes a stationary shaft on which is mounted a rotary hub to which magnetic disks can be secured. There is no direct contact between the rotating hub and the shaft. Instead, a lubricating fluid such as air or oil forms a hydrodynamic bearing between the shaft and the rotary hub. Hydrodynamic pressure or pumping is frequently provided by a pattern of grooves, commonly in a herringbone configuration, defined either by the exterior surface of the shaft or the interior surface of the rotary hub. During rotation of the hub, the pattern of grooves provides sufficient hydrodynamic pressure to cause the lubricating fluid to act as a bearing between the shaft and the rotary hub. Frequently, capillary seals are used to retain the bearing fluid between the shaft and the rotary hub.

When used in association with spindle motors, air bearings provide numerous advantages. For example, air bearings are more efficient and consume less power than either ball bearings or oil bearings. Also, air bearings are quiet and have excellent run out characteristics. Air bearings also have disadvantages. For example, when air bearings are used in disk drive spindle motors, it can be difficult or expensive to simultaneously provide both thrust (e.g. axial) and journal (e.g. radial) bearing support. Also, sliding friction associated with thrust operations during motor start-up and shut-down can create wear debris that reduces the efficiency of the motor. Additionally, air bearings often require more space than either ball or oil bearings thereby providing less space

for the motor. Finally, air bearings are typically not effective for low rotational speed applications.

Oil bearings also have advantages when applied to disk drive spindle motors. For example, oil bearings are generally quiet and have good run out characteristics. Also, oil bearings occupy less space than either ball bearings or air bearings. However, oil bearings also have disadvantages. For example, oil bearings consume more power than ball bearings or air bearings. Furthermore, when oil bearings are used in the journal bearing environment, oil leakage can be problematic.

In the future, spindle motor disk rotation speeds will steadily increase. As disk rotation speeds increase, the problems associated with standard oil bearings, air bearings and ball bearings will become magnified. Increased disk recording density is another trend in the industry. The combination of increased disk rotation speeds and increased recording densities will require disk drives to operate with improved run out characteristics. What is needed is a disk drive bearing system that has excellent run out characteristics, that demonstrates long life even when used at high rotational speeds, that eliminates oil leaks, that reduces wear and friction, that generates power savings, and that is relatively efficient to manufacture.

SUMMARY OF THE INVENTION

One aspect of the present invention relates to a spindle motor having a shaft including an outer circumferential surface. A thrust plate is fixedly connected to the shaft. The thrust plate projects radially outward from the outer circumferential surface of the shaft and includes oppositely facing top and bottom surfaces. The spindle motor also includes a hub assembly mounted on the shaft. The hub assembly is adapted for mounting a storage disk. The spindle motor further includes a liquid hydrodynamic bearing and an aerodynamic bearing. The liquid hydrodynamic bearing is formed along the top and bottom surfaces of the thrust plate and is adapted for transferring loads in an axial direction relative to the shaft. The aerodynamic bearing is formed along a portion of the hub assembly and is adapted for transferring loads in a radial direction relative to the shaft.

Another aspect of the present invention relates to a spindle motor as described above that is incorporated within a data storage system.

These and various other advantages and features of novelty that characterize the invention are pointed out with particularity in the claims annexed hereto and form a part hereof. However, for a better understanding of the invention, its advantages, and the objects obtained by its use, reference should be made to the drawings that form a further part hereof, and to accompanying descriptive matter, in which there are illustrated and described specific examples of apparatuses in accordance with the invention.

BRIEF DESCRIPTION OF THE DRAWINGS

Referring now to the drawings in which like reference numbers represent corresponding parts throughout:

FIG. **1** is a schematic diagram of a data storage system;

FIG. **2** is a top view of the system of FIG. **1**;

FIG. **3** is a cross-sectional view bisecting a first embodiment of a spindle motor constructed in accordance with the principles of the present invention;

FIG. **4** is a top plan of a thrust plate suitable for use with the spindle motor of FIG. **3**;

FIG. **5** is a perspective view of a hub sleeve suitable for use with the spindle motor of FIG. **3**;

FIG. 6 is a cross-sectional view bisecting a second embodiment of a spindle motor constructed in accordance with the principles of the present invention;

FIG. 7 is a cross-sectional view bisecting a third embodiment of a spindle motor constructed in accordance with the principles of the present invention;

FIG. 8 is a cross-sectional view bisecting a fourth embodiment of a spindle motor constructed in accordance with the principles of the present invention;

FIG. 9 is a cross-sectional view bisecting a fifth embodiment of a spindle motor constructed in accordance with the principles of the present invention; and

FIG. 10 is a cross-sectional view bisecting a sixth embodiment of a spindle motor constructed in accordance with the principles of the present invention.

DETAILED DESCRIPTION OF THE INVENTION

In the following description of the exemplary embodiments, reference is made to the accompanying drawings that form a part hereof, and in which are shown by way of illustration specific non-limiting embodiments in which the invention may be practiced. It is to be understood that other embodiments may be utilized as structural changes may be made without departing from the scope of the present invention. For ease of description, orientational or directional phrases such as "top" and "bottom" are used throughout the specification. However, it will be appreciated that the various embodiments disclosed in this specification can be used in different orientations than those specifically shown.

FIGS. **1** and **2** schematically show a data storage system **10** suitable for practicing the present invention. The system **10** includes a plurality of magnetic recording disks **12**. Each disk **12** has a plurality of concentric data tracks. The disks **12** are mounted on a spindle **14** which is connected to a spindle motor **16**. The motor **16** is mounted to a chassis **18**. The disks **12**, spindle **14**, and motor **16** form a disk stack assembly **20**.

A plurality of sliders **30** having read/write heads are positioned over the disks **12** such that each of the disks **12** has a corresponding slider **30**. Each slider **30** is attached to one of the plurality of suspensions **32** which in turn are attached to a plurality of actuator arms **34**. The arms **34** are connected to a rotary actuator **36**. Alternatively, the arms **34** may be an integral part of a rotary actuator comb. The actuator **36** moves the heads in a radial direction across the disks **12**. The actuator **36** typically comprises a rotating member **38** mounted to a rotating bearing **40**, a motor winding **42** and motor magnets **44**. The actuator **36** is also mounted to the chassis **18**. Although a rotary actuator is shown in the preferred embodiment, a linear actuator could also be used. The sliders **30**, suspensions **32**, arms **34**, and actuator **36** form an actuator assembly **46**. The disk stack assembly **20** and the actuator assembly **46** are sealed in an enclosure **48** (shown by dashed line) which provides protection from particulate contamination.

A controller unit **50** provides overall control to the system **10**. The controller unit **50** typically contains a central processing unit (CPU), memory unit and other digital circuitry. The controller **50** is connected to an actuator control/drive unit **56** which in turn is connected to the actuator **36**. This allows the controller **50** to control the movement of the sliders **30** over the disks **12**. The controller **50** is also connected to a read/write channel **58** which in turn is connected to the heads of the sliders **30**. This allows the controller **50** to send and receive data from the disks **12**. The controller **50** is further connected to a spindle control/drive unit **60** which in turn is connected to the spindle motor **16**. This allows the controller **50** to control the rotation of the disks **12**. A host system **70**, which is typically a computer system, is connected to the controller unit **50**. The host system **70** may send digital data to the controller **50** to be stored on the disks **12**, or may request that digital data be read from the disks **12** and sent to the system **70**. The basic operation of DASD units is well known in the art and is described in more detail in *Magnetic Recording Handbook* C. Dennis Mee and Eric D. Daniel, McGraw Hill Book Company, 1990.

FIG. **3** is diagramatic cross-sectional view of a spindle motor **120a** that is an embodiment of the present invention. A preferred application of the spindle motor is in data storage systems such as the disk drive system **10** illustrated in FIGS. **1** and **2**.

Referring to FIG. **3**, the spindle motor **120a** includes a shaft **122a** that is fixed relative to a base plate **124a**. A hub assembly **126a** is rotatably mounted on the shaft **122a**. The hub assembly **126a** includes a main hub **128a**. A circumferential flange **131a** projects radially outward from the main hub **128a**. The circumferential flange **131a** is adapted for securing a data storage disk **132a** (e.g., a magnetic or optical disk) to the main hub **128a**.

The shaft **122a** of the spindle motor **120a** is generally cylindrical and includes a circumferential outer surface **134a**. A top end **136a** of the shaft **122a** is positioned adjacent a cover **137a** of the spindle motor **120a**, while a bottom end **138a** of the shaft **122a** is positioned adjacent the base plate **124a**. The top and bottom ends **136a** and **138a** of the shaft **122a** are respectively connected to the cover **137a** and the base plate **124a** by fasteners such as bolts or screws **140a**.

Referring still to FIG. **3**, a generally annular thrust plate **142a** is fixably connected to the shaft **122a**. As shown in FIG. **3**, the shaft **122a** and the thrust plate **142a** are constructed from different components. However, it will be appreciated that in alternative embodiments, the shaft **122a** and the thrust plate **142a** can be unitarily constructed as a single integral piece.

The thrust plate **142a** projects radially outward from the circumferential outer surface **134a** of the shaft **122a**. The thrust plate **142a** includes substantially parallel top and bottom sides **146a** and **148a**. The thrust plate **142a** is mounted within a first annular recess **150a** defined within the top of the main hub **128a**. As mounted in the first annular recess **150a**, the bottom side **148a** opposes a first radial shoulder **152a** of the main hub **128a**.

A containment plate **154a** is mounted on the shaft **122a** and is fixedly connected to the main hub **128a**. The containment plate **154a** is generally annular and the shaft **122a** extends through a central opening of the containment plate **154a**. The containment plate **154a** is positioned directly above the thrust plate **142a** and includes a fluid containment surface **156a** that faces downwardly and opposes the top side **146a** of the thrust plate **142a**. The containment plate **154a** is mounted in a second annular recess **155a** defined in the top of the main hub **128a**. When mounted within the second annular recess **155a**, the fluid containment surface **156a** of the containment plate **154a** abuts against a second radial shoulder **157a** defined by the main hub **128a**.

The spindle motor **120a** includes a liquid hydrodynamic bearing for supporting load applied in an axial direction relative to the shaft **122a**. For example, a liquid such as grease or oil **158a** is filled between the thrust plate **142a**, the

main hub **128a** and the containment plate **154a**. Preferably, a spacing gap of about 10 microns exists between the thrust plate **142a** and the main hub **128a**, and between the thrust plate **142a** and the containment plate **154a**. Porting holes **160a** are defined through the thrust plate **142a** for allowing air to escape when the oil **158a** is pressurized.

The spindle motor **120a** preferably includes structure for pressurizing the oil **158a** when the main hub **128a** is rotated relative to the shaft **122a**. For example, the top and bottom sides **146a** and **148a** of the thrust plate **142a** can define a pattern of grooves adapted for pumping the oil **158a** radially outward relative to the shaft **122a** when the main hub **128a** is rotated. An exemplary herringbone pattern **162a** is shown in FIG. **4**. It will be appreciated that similar patterns can also be formed on the fluid containment surface **156a** of the containment plate **154a** and the first radial shoulder **152a** of the main hub **128a** to achieve a similar result.

As described below, the spindle motor **120a** preferably uses an aerodynamic bearing as a radial or journal bearing. The radial bearing is adapted for transferring loads in a radial direction relative to the shaft **122a**.

Referring again to FIG. **3**, the hub assembly **126a** includes a hub sleeve **162a** connected to the main hub **128a** by fasteners **164a** such as bolts, screws or adhesives. The hub sleeve **162a** is generally annular and is concentrically mounted with respect to the shaft **122a**. A stationary sleeve **166a** is mounted directly outside the hub sleeve **162a**. The stationary sleeve **166a** is generally annular and is also generally concentric with respect to the shaft **122a**. A lower portion of the stationary sleeve **166a** is fixedly connected to a support member **168a** that is fixedly connected to the shaft **122a**. The stationary sleeve **166a** is preferably mounted between the flange **131a** and the hub sleeve **162a**.

An aerodynamic bearing is preferably formed between the hub sleeve **162a** and the stationary sleeve **166a**. A preferred spacing gap between the hub sleeve **162a** and the stationary sleeve **166a** is about 2 microns. The spindle motor **120a** preferably includes structure for generating air pressure between the hub sleeve **162a** and the stationary sleeve **166a** such that the air bearing is generated. For example, a pattern of grooves configured for pumping air can be defined by either the outer surface of the hub sleeve **162a** or the inner surface of the stationary sleeve **166a**.

FIG. **5** shows one particular groove pattern that is formed on the outer surface of the hub sleeve **162a**. The groove pattern of FIG. **5** includes an upper set of grooves **170a** and a lower set of grooves **172a**. The upper and lower sets of grooves **170a** and **172a** are arranged generally in a herringbone configuration. The upper set of grooves **170a** is adapted to pump air in a downward direction while the lower set of grooves **172a** is adapted to pump air in an upward direction. The lengths of the upper and lower sets of grooves **170a** and **172a** are selected to make the upward and downward pumping tendency equal or balanced.

Referring back to FIG. **3**, the spindle motor **120a** further includes a generally annular magnet **174a** secured to the inside of the hub sleeve **162a**. The magnet **174a** surrounds a stator **176a** that is fixed to the shaft **122a**.

In use of the spindle motor **120a**, load in an axial direction relative to the shaft **122a** is supported by the liquid hydrodynamic bearing formed between the containment plate **154a**, the thrust plate **142a** and the main hub **128a**. In contrast, load in a radial direction relative to the shaft **122a** is supported by the aerodynamic bearing formed between the hub sleeve **162a** of the hub assembly **126a** and the stationary sleeve **166a** secured to the shaft **122a**.

FIG. **6** illustrates a spindle motor **120b** that is a second embodiment of the present invention. The spindle motor **120b** includes a fixed shaft **122b** and a main hub **128b** rotatably mounted on the fixed shaft **122b**. The spindle motor **120b** also includes a containment plate **154b** and a thrust plate **142b** that cooperate with the main hub **128b** to form a liquid hydrodynamic bearing that is identical to the liquid hydrodynamic bearing described with respect to the spindle motor **120a** of FIG. **3**.

The spindle motor **120b** has been modified to include a different aerodynamic bearing configuration as compared to the spindle motor **120a** of FIG. **3**. For example, the spindle motor **120b** includes a stationary sleeve **166b** that is generally annular and is fixedly connected directly to the shaft **122b**. An annular hub sleeve **162b** is fixedly connected to the main hub **128b**. The hub sleeve **162b** is positioned outside the stationary sleeve **166b**. Both the stationary sleeve **166b** and the hub sleeve **162b** are concentric with respect to the shaft **122b**. An annular magnet **174b** is connected to an outer surface of the hub sleeve **162b**. The magnet **174b** is surrounded by an annular stator **176b** that is connected to a base plate **124b** of the spindle motor **120b**.

Either the outer surface of the stationary sleeve **166b** or the inner surface of the hub sleeve **162b** defines a pattern of grooves adapted for pumping air between the hub sleeve **162b** and the stationary sleeve **166b** when the main hub **128b** is rotated relative to the shaft **122b**. The pumping action is adapted for generating an aerodynamic bearing between the hub sleeve **162b** and the stationary sleeve **166b**.

FIG. **7** illustrates a spindle motor **120c** having a modified thrust bearing configuration as compared to the spindle motor **120a** of FIG. **3**. The spindle motor **120c** includes a fixed shaft **122c** on which a main hub **128c** is rotatably mounted. The shaft **122c** includes a bottom end **138c** connected to a base plate **124c**, and a top end **136c** positioned directly beneath a containment plate **154c**. The containment plate **154c** has a generally solid, disk-shaped configuration and is fixedly connected to the main hub **128c**. A fluid containment surface **156c** of the containment plate **154c** faces the top end **136c** of the shaft **122c**. A thrust plate **142c** is fixedly connected to the top end **136c** of the shaft **122c**. The thrust plate **142c** projects radially outward from the shaft **122c** and includes a top side **146c** that faces the fluid containment surface **156c** of the containment plate **154c**, and a bottom side **148c** that faces a radial shoulder **152c** of the main hub **128c**. A lubricant such as oil or grease is filled between the containment plate **154c**, the thrust plate **142c** and the main hub **128c**. In certain embodiments, the top and bottom sides **146c** and **148c** of the thrust plate **142c** define a pattern of grooves configured for pressurizing the lubricant when the main hub **128c** is rotated relative to the shaft **122c**. In other embodiments, a similar groove configuration can be formed on the containment plate **154c** and the radial shoulder **152c** to achieve a similar result. In this manner, the main hub **128c**, the thrust plate **142c**, the containment plate **154c** and the lubricant cooperate to provide a liquid hydrodynamic thrust bearing.

FIG. **8** shows a spindle motor **120d** that is a fourth embodiment of the present invention. The spindle motor **120d** includes a shaft **122d**, a thrust plate **142d** and a containment plate **154d** that are arranged in the same configuration as the shaft **122c**, the thrust plate **142c** and the containment plate **154c** of FIG. **7**. Additionally, the spindle motor **120d** includes a main hub **128d**, a stationary sleeve **166d**, a hub sleeve **162d**, a magnet **174d** and a stator **176d** that are arranged in the same configuration as the corresponding components of the spindle motor **120b** of FIG. **6**.

FIG. 9 illustrates a fifth spindle motor **120e** constructed in accordance with the principles of the present invention. The spindle motor **120e** includes a rotatable shaft **122e** that is free to rotate relative to a base plate **124e**. A main hub **128e** is fixedly connected to a top end of the shaft **122e**. A flange **131e** for mounting a storage disk **132e** projects radially outward from the main hub **128e**.

A stationary sleeve **166e** is fixedly connected to the base plate **124e** by suitable fasteners **163e**. The stationary sleeve **166e** is generally concentric with respect to the shaft **122e**. A bottom side of the stationary sleeve **166e** defines a first annular recess **150e** for receiving a thrust plate **142e**, and a second annular recess **155e** for receiving a containment plate **154e**. The thrust plate **142e** is fixedly connected to a bottom end **138e** of the shaft **122e**. The thrust plate **142e** projects radially outward from an outer circumferential surface **134e** of the shaft **122e** and includes substantially parallel top and bottom sides **146e** and **148e**. The top side **146e** of the thrust plate **142e** faces a first radial shoulder **152e** of the stationary sleeve **166e**. The bottom side **148e** of the thrust plate **142e** faces a fluid containment surface **156e** of the containment plate **154e**. The containment plate **154e** is fixedly connected to the stationary sleeve **166e**, and the fluid containment surface **156e** engages a second radial shoulder **157e** of the stationary sleeve **166e**. A liquid lubricant such as oil or grease is filled between the stationary sleeve **166e**, the thrust plate **142e** and the containment plate **154e**. When the shaft **122e** is rotated, the lubricant is pressurized (e.g., by a pattern of grooves as previously described) to provide a liquid hydrodynamic bearing for transferring loads in an axial direction relative to the shaft **122e**.

The spindle motor **120e** also includes a hub sleeve **162e** fixedly connected to the main hub **128e**. The hub sleeve **162e** is generally annular and includes an inner surface that faces a corresponding outer surface of the stationary sleeve **166e**. The hub sleeve **162e** and the stationary sleeve **166e** are configured to provide an air journal bearing when the hub **128e** is rotated. For example, either the inner surface of the hub sleeve **162e** or the outer surface of the stationary sleeve **166e** can have a herringbone pattern of grooves for pumping air as previously described in the specification. The spindle motor **120e** further includes an annular magnet **174e** fastened to the shaft **122e**, and a stator **176e** that surrounds the magnet **174e**. The magnet **174e** and the stator **176e** are positioned within an inner chamber defined by the stationary sleeve **166e**. The stator **176e** is fixedly connected to the sleeve **166e**.

FIG. **10** illustrates a sixth embodiment of a spindle motor **120f** constructed in accordance with the principles of the present invention. The spindle motor **120f** has substantially the same thrust bearing configuration as the liquid hydrodynamic thrust bearing described with respect to FIG. **9**. However, the spindle motor **120f** of FIG. **10** includes a stationary sleeve **166f** that does not have an internal chamber for housing a stator or a magnet. Instead, a magnet **174f** is mounted on the outside of a hub sleeve **162f**. Additionally, a stator **176f** is mounted outside the magnet **174f**. Preferably, an air journal bearing is formed between an inner surface of the hub sleeve **162f** and an outer surface of the stationary sleeve **166f**.

The various aspects of the present invention provide numerous advantages. For example, the use of an air journal bearing provides reduced motor drag as compared to spindle motors having oil journal bearings. The use of air journal bearings also eliminates the oil leakage problem commonly associated with oil journal bearings. Furthermore, the use of a liquid thrust bearing assists in reducing frictional wear associated with starting and stopping operations, and also eliminates the precise tolerances associated with air thrust bearings. Moreover, the various embodiments of the present invention can be efficiently and cost effectively manufactured. Additionally, the configurations of the thrust and journal bearings provide for spindle motors that can be efficiently operated at various different orientations. For example, the embodiments can be effectively operated right-side-up, upside-down, or angled relative to horizontal.

The foregoing description of the exemplary embodiment of the invention has been presented for the purposes of illustration and description. It is not intended to be exhaustive or to limit the invention to the precise form disclosed. Many modifications and variations are possible in light of the above teaching. It is intended that the scope of the invention be limited not with this detailed description, but rather by the claims appended hereto.

What is claimed is:

1. A spindle motor comprising:

a shaft having an outer circumferential surface;

a stationary sleeve concentrically positioned about the shaft;

a hub assembly including a hub mounted on the shaft, the hub being rotatable relative to the stationary sleeve, the hub being adapted for mounting a storage disc, and the hub assembly including a hub sleeve connected to the hub;

a thrust plate fixedly connected to the shaft, the thrust plate projecting radially outward from the outer circumferential surface of the shaft;

a liquid hydrodynamic bearing formed adjacent the thrust plate for transferring loads in an axial direction relative to the shaft; and

an aerodynamic bearing formed between the stationary sleeve and the hub sleeve for transferring loads in a radial direction relative to the shaft.

2. The spindle motor of claim **1**, further comprising a base to which the shaft is fixedly connected, and a containment plate connected to the hub, the containment plate having a fluid containment surface positioned adjacent to the thrust plate, wherein the liquid hydrodynamic bearing is formed by a lubricating liquid positioned between the thrust plate and hub and also between the thrust plate and the fluid containment surface of the containment plate.

3. The spindle motor of claim **2**, wherein the lubricating liquid comprises oil or grease.

4. The spindle motor of claim **2**, wherein the hub defines a radial shoulder on which the thrust plate is seated.

5. The spindle motor of claim **4**, wherein the thrust plate is captured between the shoulder of the hub and the fluid containment surface of the containment plate.

6. The spindle motor of claim **5**, wherein patterns of grooves are formed on one of a) top and bottom surfaces of the thrust plate and b) the containment surface of the containment plate and the shoulder of the hub, for pressurizing the lubricating liquid when the shaft is rotated such that the pressurized liquid provides the liquid hydrodynamic bearing.

7. The spindle motor of claim **1**, wherein the hub sleeve is mounted inside the stationary sleeve, and a pattern of grooves is formed on one of an outer surface of the hub sleeve and an inner surface of the stationary sleeve, the pattern of grooves being configured for pressurizing air between the stationary sleeve and the hub sleeve when the hub is rotated, wherein the pressurized air provides the aerodynamic bearing.

8. The spindle motor of claim 7, wherein a magnet is secured to an inside of the hub sleeve, and a stator is secured between the magnet and the shaft.

9. The spindle motor of claim 5, wherein a bearing gap of around 10 microns exists between the thrust plate and the recessed shoulder, and between the thrust plate and the fluid containment surface of the containment plate.

10. The spindle motor of claim 1, wherein a bearing gap of around 2 microns exists between the stationary sleeve and the hub sleeve.

11. The spindle motor of claim 1, wherein the hub sleeve is mounted outside the stationary sleeve, and a pattern of grooves is formed on one of an inner surface of the hub sleeve and an outer surface of the stationary sleeve, the pattern of grooves being configured for pressurizing air between the stationary sleeve and the hub sleeve when the hub is rotated, wherein the pressurized air provides the aerodynamic bearing.

12. The spindle motor of claim 11, wherein a magnet is secured to an outside of the hub sleeve, and a stator is secured outside the magnet.

13. The spindle motor of claim 11, wherein a stator is secured to an inside of the hub sleeve, and a magnet is positioned between the stator and the shaft.

14. The spindle motor of claim 1, wherein the hub is fixedly connected to the shaft, and the motor further includes a containment plate connected to the stationary sleeve, the containment plate having a fluid containment surface positioned adjacent to the thrust plate, wherein the liquid hydrodynamic bearing is formed by a lubricating liquid positioned between the thrust plate and stationary sleeve and also between the thrust plate and the fluid containment surface of the containment plate.

15. The spindle motor of claim 14, wherein the lubricating liquid comprises oil or grease.

16. The spindle motor of claim 14, wherein the stationary sleeve defines a radial shoulder adapted to oppose the thrust plate.

17. The spindle motor of claim 16, wherein the thrust plate is captured between the radial shoulder of the stationary sleeve and the fluid containment surface of the containment plate.

18. The spindle motor of claim 17, wherein patterns of grooves are formed on one of a) top and bottom surfaces of the thrust plate and b) the containment surface of the containment plate and the radial shoulder of the stationary sleeve, for pressurizing the lubricating liquid when the shaft is rotated such that the pressurized liquid provides the liquid hydrodynamic bearing.

19. A spindle motor comprising:

a shaft having an outer circumferential surface;

a thrust plate fixedly connected to the shaft that projects radially outward from the outer circumferential surface of the shaft, the thrust plate having top and bottom surfaces;

a hub assembly mounted on the shaft, the hub assembly being adapted for mounting a storage disc;

a liquid hydrodynamic bearing formed along the top and bottom surfaces of the thrust plate for transferring loads in an axial direction relative to the shaft; and

an aerodynamic bearing formed along a portion of the hub assembly for transferring loads in a radial direction relative to the shaft.

20. The spindle motor of claim 19, further comprising a stationary sleeve positioned inside a hub of the hub assembly, wherein aerodynamic bearing is formed between the stationary sleeve and hub.

21. A data storage device comprising:

a spindle motor including:

a shaft having an outer circumferential surface;

a thrust plate fixedly connected to the shaft that projects radially outward from the outer circumferential surface of the shaft, the thrust plate having top and bottom surfaces;

a hub assembly mounted on the shaft, the rotor structure being adapted for mounting a storage disc;

a liquid hydrodynamic bearing formed along the top and bottom surfaces of the thrust plate for transferring loads in an axial direction relative to the shaft; and

an aerodynamic bearing formed along a portion of the hub assembly for transferring loads in a radial direction relative to the shaft; and

a data storage medium coupled to the rotor structure;

a transducer for reading from and writing to the data storage medium; and

an actuator assembly for moving the transducer relative to the data storage medium.

* * * * *

US005540504A

United States Patent [19]

C rdova et al.

[11] Patent Number: 5,540,504

[45] Date of Patent: Jul. 30, 1996

[54] SHOCK RESISTANT BEARING

[75] Inventors: Jackie Cordova; Richard E. Mills, both of Colorado Springs, Colo.

[73] Assignee: Quantum Corporation, Milpitas, Calif.

[21] Appl. No.: 524,716

[22] Filed: Sep. 7, 1995

[51] Int. Cl.[6] F16C 32/06

[52] U.S. Cl. 384/100; 384/107; 384/292; 384/378

[58] Field of Search 384/100, 107, 384/111, 113, 397, 378, 292

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5,127,744	7/1992	White et al.	384/107 X
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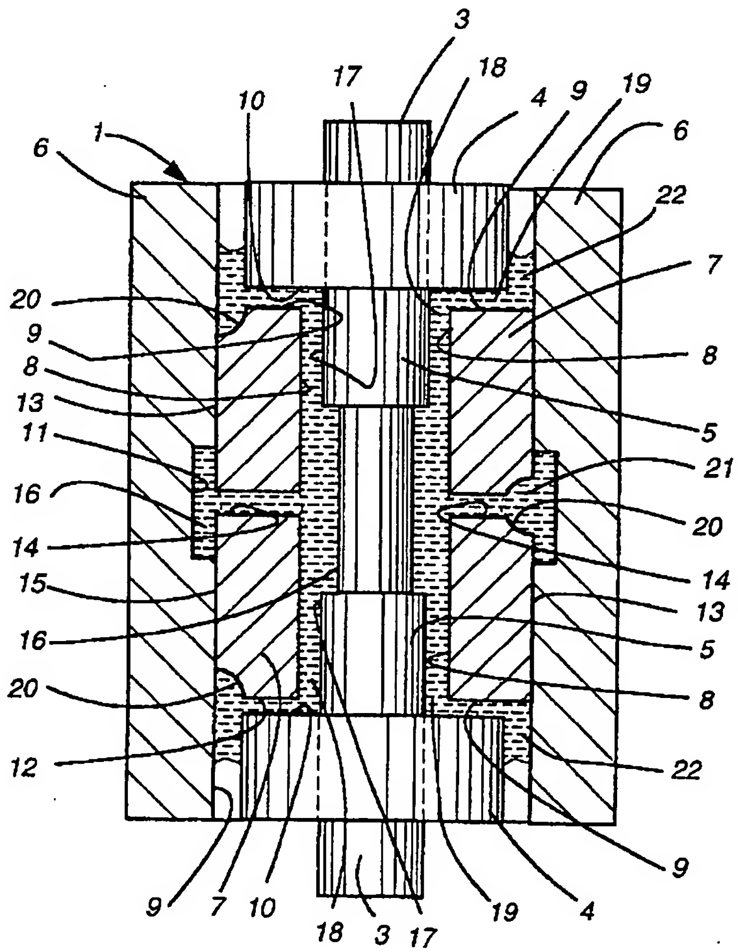
Primary Examiner—Thomas R. Hannon
Attorney, Agent, or Firm—David B. Harrison

[57] ABSTRACT

A hydrodynamic bearing journaling a shaft includes a rotatable bushing journaling the shaft and thrust plates on the shaft at opposite ends of the bushing. The bushing is encased in a cylindrical sleeve. Lubricant containing clearance spaces are formed between the shaft, bushing, sleeve, and thrust plates. One or more internal helical passages are formed between the bushing and sleeve and open into the axial ends thereof adjacent to the thrust plates. An internal radial chamber between the bushing and sleeve opens into the helical channel through radial ports for providing lubricant between the journal bearing and the shaft and thrust plates.

8 Claims, 3 Drawing Sheets



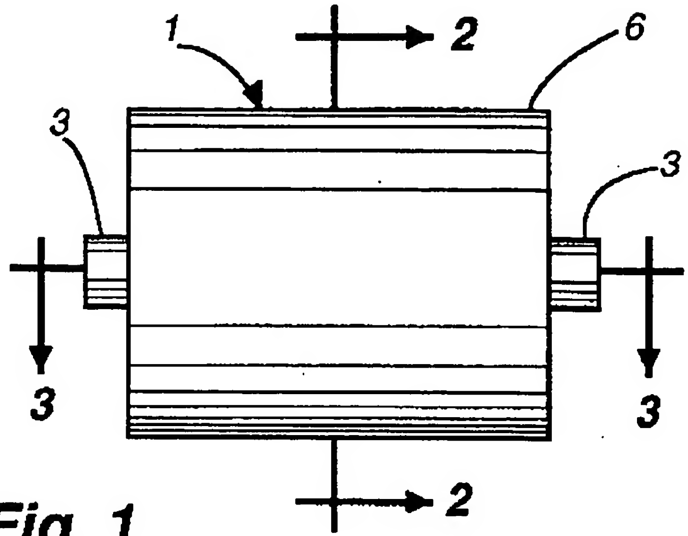


Fig. 1

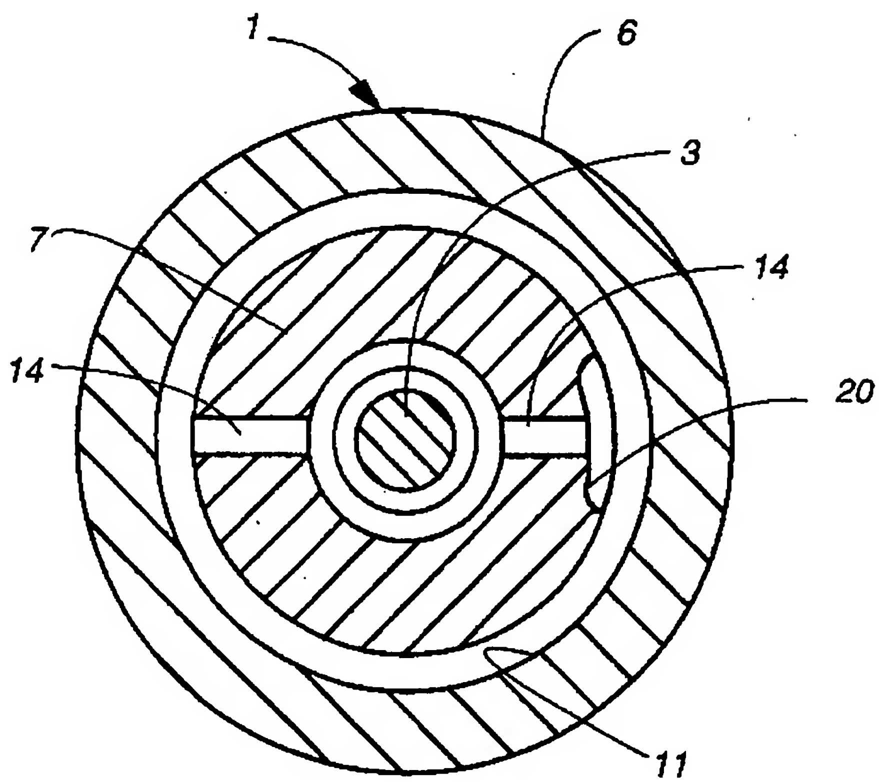


Fig. 2

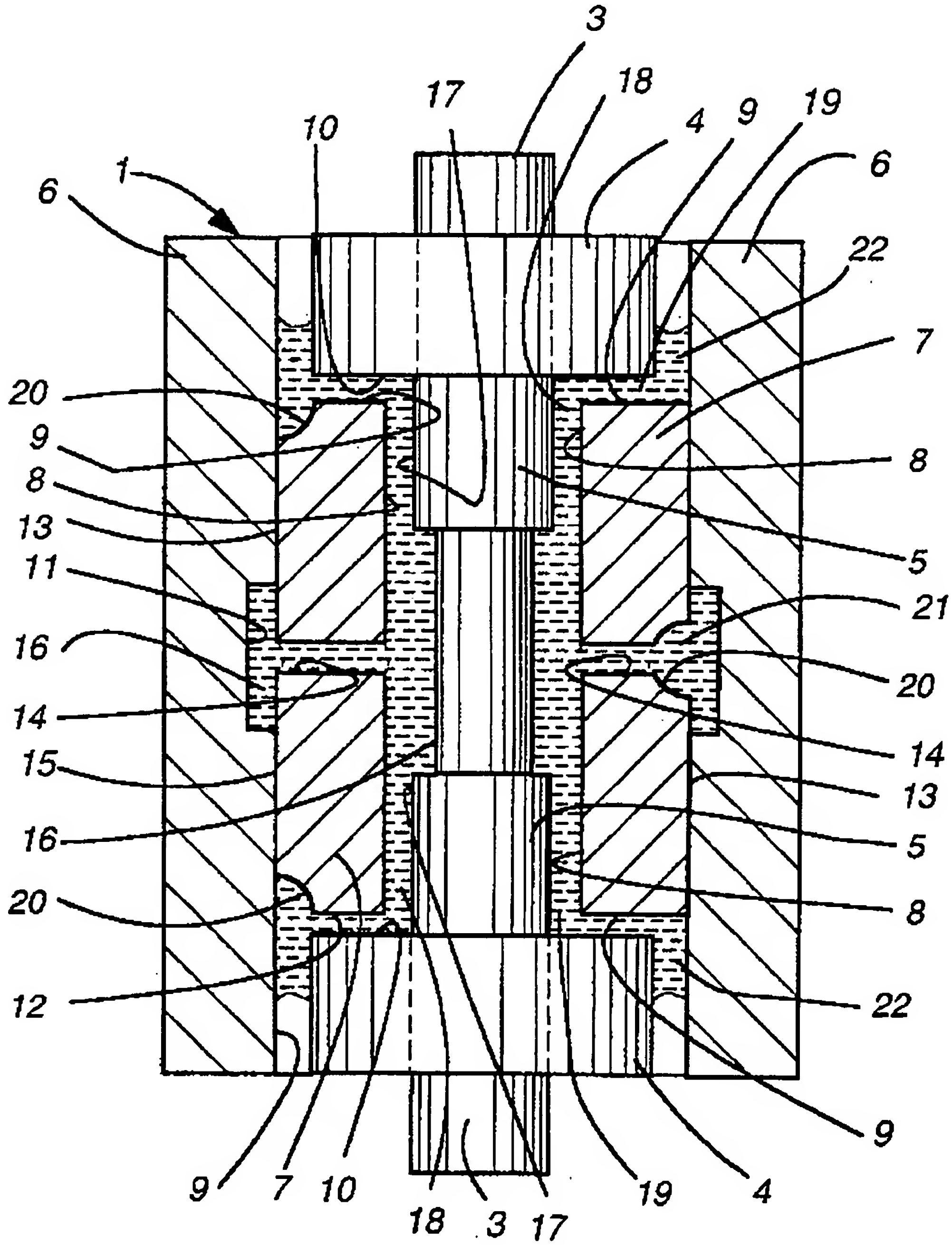

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Fig. 3

Fig. 4

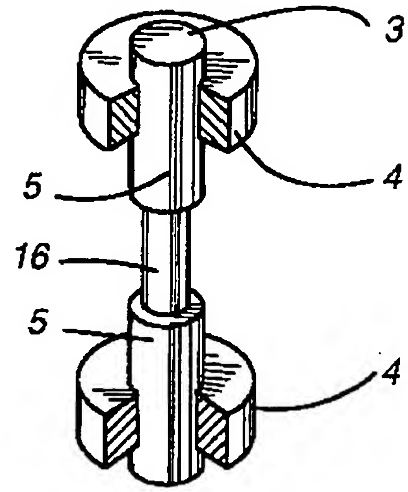


Fig. 5

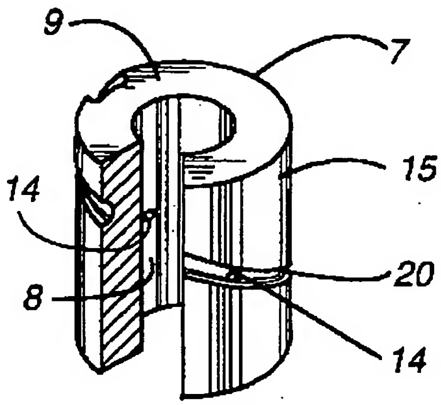
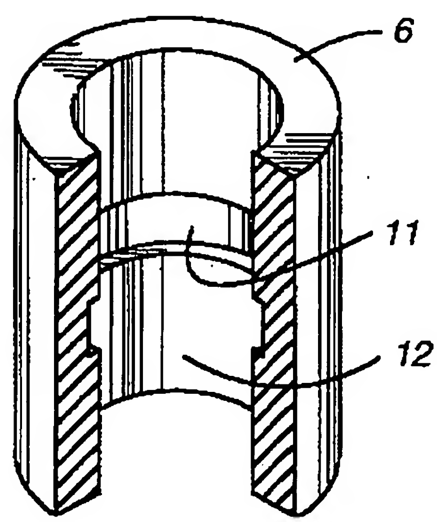


Fig. 6



SHOCK RESISTANT BEARING

BACKGROUND OF THE INVENTION

1. Field of the Invention

The present invention relates to precision journal bearings for high speed precision drive shafts. More particularly the present invention relates to high speed, internally lubricated, hydrodynamic journal or spindle bearings finding particular but not necessarily exclusive utility for magnetic media drive mechanisms.

2. Description of the Prior Art

Bearings of the character embodying the present invention are disclosed in U.S. Pat. No. 4,795,275, issued Jan. 3, 1989, to F. Titcomb and J. Cordova, for "Hydrodynamic Bearing." Such bearings are also disclosed in U.S. Pat. No. 4,596,474, issued Jun. 24, 1986, to F. Van Roemburg; U.S. Pat. No. 4,726,693, issued Feb. 23, 1988, to J. Anderson and R. Sleger; U.S. Pat. No. 5,067,528, issued Nov. 26, 1991, to F. Titcomb and J. Cordova; and U.S. Pat. No. 5,112,142, issued May 12, 1992, to F. Titcomb and J. Cordova. Such hydrodynamic bearings include a bearing sleeve with an internal journal bushing press fitted therein and in which is journaled a precision shaft, with provision between the shaft and bushing for incorporating lubricants. Thrust bearings are mounted on the shaft on opposite sides of the bushing. Flats are conventionally machined on the exterior surface of the bushing before press or shrink fit assembly into the sleeve to provide axially extending pressure equalizing lubricant passages in the bearing, with ports or passages through the bushing for conducting lubricant to the shaft journaled in the bushing and to the thrust bearings.

OBJECTS OF THE INVENTION

It is the principal object of the present invention to provide an improved hydrodynamic bearing.

More specifically, it is an object of the present invention to provide an improved hydrodynamic bearing having enhanced shock resistance.

It is another object of the present invention to provide an improved hydrodynamic, internally lubricated shaft and journal sleeve bearing structure having enhanced shock resistance, which is readily constructed and assembled, rugged in use, and suitable for high speed equipment such as magnetic media disk drives.

SUMMARY OF THE INVENTION

In accordance with the foregoing objects, the present invention is embodied in a hydrodynamic bearing comprising an outer cylindrical sleeve having a bushing with a smaller inside diameter press or shrink fitted therein and a shaft having journals rotatably journaled in the bushing. A pair of thrust plates are mounted on the shaft and rotatably and sealingly coact with radially extending faces on the smaller diameter portion of the bushing. The clearance spaces between the bushing, shaft journals and thrust plates are filled with a lubricant. The external faces of the thrust plates are exposed to the air. The larger diameter portion of the bushing includes one or more helical axially extending passageways which, with radial ports through the bushing, define a lubricant chamber which retains lubricant in the event of a jar or shock to the bearing.

DESCRIPTION OF THE DRAWINGS

FIG. 1 is an elevation view of a hydrodynamic bearing embodying the present invention.

FIG. 2 is a vertical section view taken substantially in the plane of line 2—2 on FIG. 1.

FIG. 3 is a horizontal section view taken substantially in the plane of line 3—3 on FIG. 1.

FIG. 4 is a perspective view of a journal bearing embodying the present invention, with parts cut away for clarity.

FIG. 5 is a perspective view of a sleeve portion of the bearing shown in FIG. 4 with, parts cut away for clarity.

FIG. 6 is a perspective view of a bushing portion of the bearing shown in FIG. 4 with, parts cut away for clarity.

DESCRIPTION OF THE PREFERRED EMBODIMENT

The present invention is embodied in an internally lubricated journal bearing **1** rotatably journaling a shaft **3** having thrust plates **4** supported on opposite ends thereof. The shaft defines spaced spools or journals **5**. The thrust plates **4** are mounted on the shaft **3** adjacent to each shaft spool or journal **5**. The thrust plates **4** may be press fit onto the shaft **3** to provide a lubricant tight seal therebetween.

The journal bearing **1** is formed by an external sleeve **6** having an internal bushing **7** press or shrink fit mounted therein. The internal bushing **7** defines an inner, axially extending journal bearing surface **8** within which the shaft journals **5** are rotatingly supported or journaled. At its opposite ends, the internal bushing **7** defines radially extending thrust surfaces **9** in opposing relationship with internal radial surfaces **10** on the thrust plates **4**.

For lubricant supply within the bearing, an internal lubricant reservoir in the form of an internal circumferential channel **11** is defined in the inner surface **12** of the outer sleeve **6** or optionally on the external surface **13** of the bushing **7**. A plurality of radial ports **14** extend from the internal or inner surface **8** of the bushing **7** to the outer surface **13** thereof and open into the lubricant reservoir channel **11** defined in the outer sleeve **6**, thereby providing a lubricant passage from the reservoir channel **11** to the shaft journals **5** and inner bushing surface **8**.

Lubricant is also contained in a circumferential recess or channel **16** which extends axially between the shaft journals **5**. The shaft journals **5** define axially extending circumferential bearing surfaces **17** which operatively coact with the internal surface **8** of the bushing and the lubricant to journal the shaft **3** in the journal bearing bushing **7**. Lubricant is contained in the journal bearing structure in the clearance chamber **18** formed and defined between the internal surface **8** of the bushing **7** and the external surface **17** of each shaft journal **5**, which chamber **18** includes the shaft recess **16** and the clearance spaces **19** defined between the facing thrust surfaces **9** and **10** of the bushing **7** and thrust plate **4**, respectively. Lubricant is supplied to the internal chamber **18** thus defined from the reservoir channel **11** defined in the sleeve, through the radial ports **14** in the bushing **7**.

For providing a further lubricant supply to provide an enhanced shock resistance to the bearing structure, the external surface **13** of the bushing **7** defines one or more spiral, helical or zigzag grooves **20** which in turn form a spiral or helical chamber or chambers **21** with the external sleeve **6** when the bushing **7** is press or shrink fitted therein. While it is preferred that the radial ports **14** do not intersect the spiral or helical grooves **20**, such ports may if desired

intersect one or more of the grooves **20**. The grooves **20** may be formed of any configuration, such as triangular, semicircular or rectangular or combinations thereof. The helical grooves **20** open into the end surfaces **9** of the bushing **7** and capillary lubricant seals **22** defined between the sleeve, bushing and thrust plates as shown in FIG. **3**, thereby providing lubricant to the spaces between the bushing end surfaces **9** and the radial surfaces **10** of the thrust plates **4**. Because of the length and volume of the helical groove or grooves, in the event of a shock or the dropping of the bearing structure, lubricant will be retained in the groove chamber thus formed by the inertia of the fluid in the groove **20**. Such a structure provides utility, for example, in a disk drive construction for use in a portable or lap top computer which is subject to relatively rough handling, dropping or shock.

The foregoing structure is produced by forming or machining the spiral or helical or zigzag groove or grooves **20** in the bushing **7** or optionally in the sleeve **6** by an appropriate machining process prior to force or shrink fitting the bushing **7** within the outer sleeve **6**, thus facilitating the precision manufacture of the bearing and shaft structure. As a result, a simple yet rugged, internally lubricated journal bearing and shaft mechanism is readily produced.

While a certain illustrative embodiment of the present invention has been shown in the drawings and described above in considerable detail, it should be understood that there is no intention to limit the invention to the specific modification or embodiment disclosed. On the contrary, the intention is to cover all modifications, alternative constructions, equivalents and uses falling within the spirit and scope of the invention as expressed in the appended claims.

We claim:

1. A hydrodynamic bearing for journaling a shaft, comprising a bushing supported in a sleeve and journaling said shaft, thrust plates on said shaft at opposite ends of said bushing, said bushing, shaft and thrust plates defining a

clearance space therebetween for containing a lubricant, characterized by an internal helical channel defined between said bushing and said sleeve and opening at the axial ends thereof to the clearance space between said thrust plates and said bushing, and a port through said bushing opening into said helical channel for providing lubricant flow through said journal bearing and said clearance space.

2. A bearing as defined in claim **1** wherein said bushing and sleeve define a plurality of internal helical channels.

3. A hydrodynamic bearing and shaft assembly comprising a cylindrical bushing encased in a cylindrical sleeve, a shaft journaled in said bushing and including a journal having an outer surface forming a first clearance space with respect to the smaller inner surface of said bushing, a pair of thrust plates mounted on said shaft and having thrust surfaces forming second clearance spaces with respect to the radially extending surfaces of said bushing and the inner surface of said thrust plate, the external faces of said thrust plates being exposed to the air, said clearance spaces being filled with a lubricant, said bushing and sleeve defining a helical, axially extending passageway therebetween communicating with said first and second clearance spaces through radial ports defined in said bushing.

4. A bearing as defined in claim **3** further including an interior circumferential channel defined between said bushing and said sleeve, and said radial ports in the bushing connecting said channel to said clearance space between said shaft and said bearing bushing.

5. A bearing as defined in claim **3** wherein said channel is defined in said sleeve.

6. A bearing as defined in claim **3** wherein said channel is defined in said bushing.

7. A bearing as defined in claim **3** wherein said channel is defined in both said sleeve and said bushing.

8. A bearing as defined in claim **3** further comprising a plurality of said helical passageways.

* * * * *

US005634724A

United States Patent [19]

Zang et al.

[11] Patent Number: 5,634,724
[45] Date of Patent: Jun. 3, 1997

[54] HYDRODYNAMIC BEARING FOR SPINDLE MOTOR HAVING HIGH INERTIAL LOAD

[75] Inventors: Yan Zang, Milpitas; Michael R. Hatch, Mountain View, both of Calif.

[73] Assignee: Quantum Corporation, Milpitas, Calif.

[21] Appl. No.: 519,842

[22] Filed: Aug. 25, 1995

[51] Int. Cl.[6] F16C 17/10

[52] U.S. Cl. 384/107; 384/124

[58] Field of Search 384/107, 112, 384/119, 124, 115, 123

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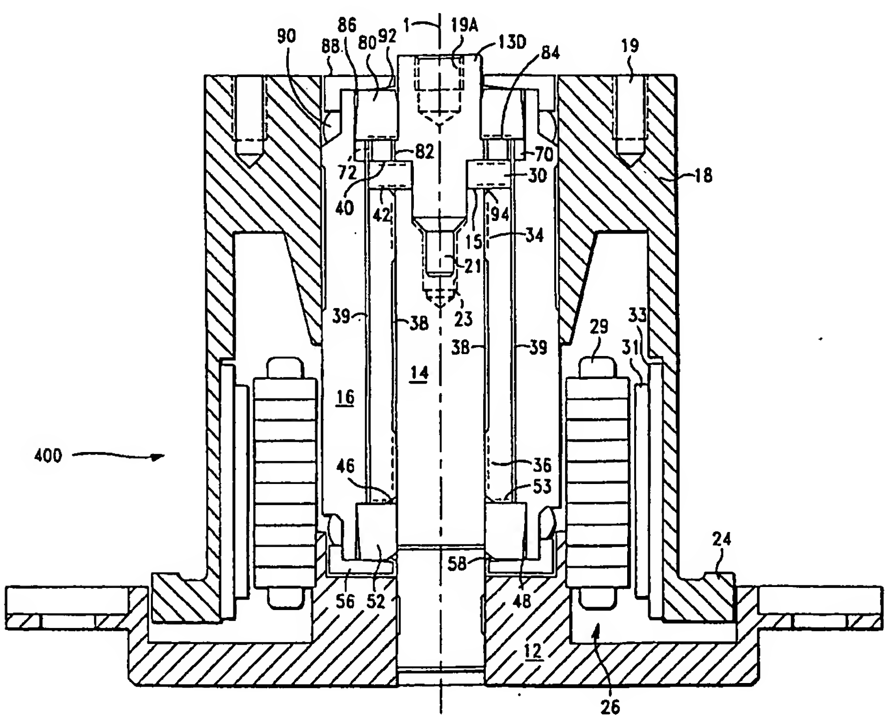
Primary Examiner—Lenard A. Footland
Attorney, Agent, or Firm—David B. Harrison

[57] ABSTRACT

A self-contained hydrodynamic bearing unit includes a shaft having a threaded axial opening at one end thereof and a shaft shoulder adjacent the opening and perpendicular with a longitudinal axis of the shaft, a sleeve defining an opening for receiving the shaft for relative rotation, a pair of longitudinally spaced-apart radial hydrodynamic journal bearings defined between the shaft and the sleeve, a shaft-bolt including a threaded end region for mating with the threaded axial opening of the shaft and defining a bolt shoulder, an annular thrust plate having two parallel radial faces and adapted to fit upon the shaft-bolt for mounting between the shaft shoulder and the shaft bolt shoulder when the shaft-bolt is mated with the threaded axial opening of the shaft, the sleeve defining a radial thrust bearing surface portion extending radially outwardly at one end and positioned to confront one radial face of the thrust plate, an annular thrust bushing mounted to the sleeve and defining a radial thrust bearing surface portion confronting the other radial face of the thrust plate, the annular thrust plate defining two axial hydrodynamic thrust bearings respectively with the radial thrust bearing surface portion of the sleeve and the radial thrust bearing surface portion of the annular thrust bushing and a hydrodynamic thrust bearing lubricant reservoir between the two axial hydrodynamic thrust bearings and between a cylindrical face of the thrust plate and a facing cylindrical wall of the sleeve, and a hydrodynamic bearing lubricant in the bearing unit and at the pair of hydrodynamic journal bearings and at the two axial hydrodynamic thrust bearings and in the hydrodynamic bearing lubricant reservoir.

30 Claims, 5 Drawing Sheets



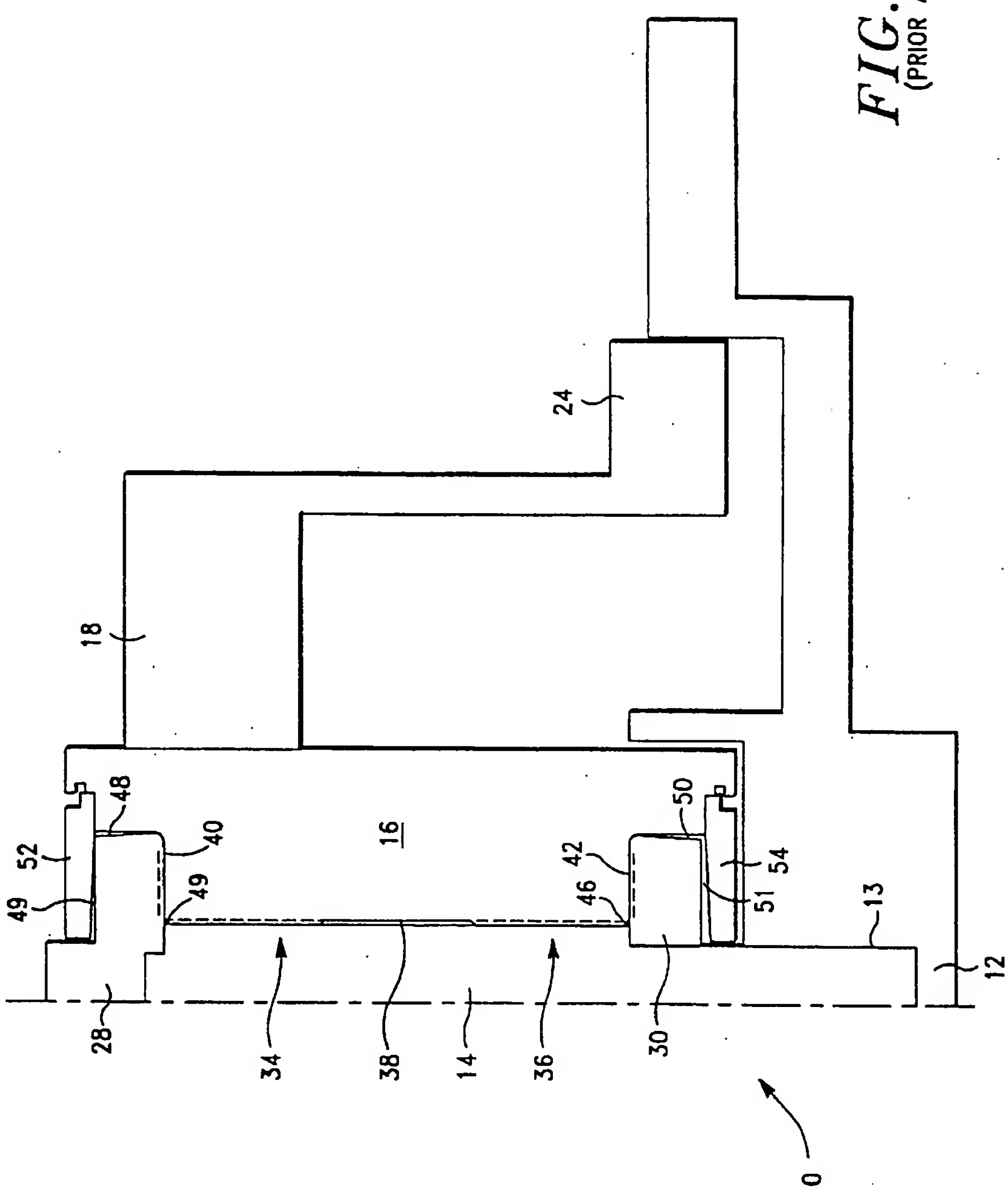


FIG.—1
(PRIOR ART)

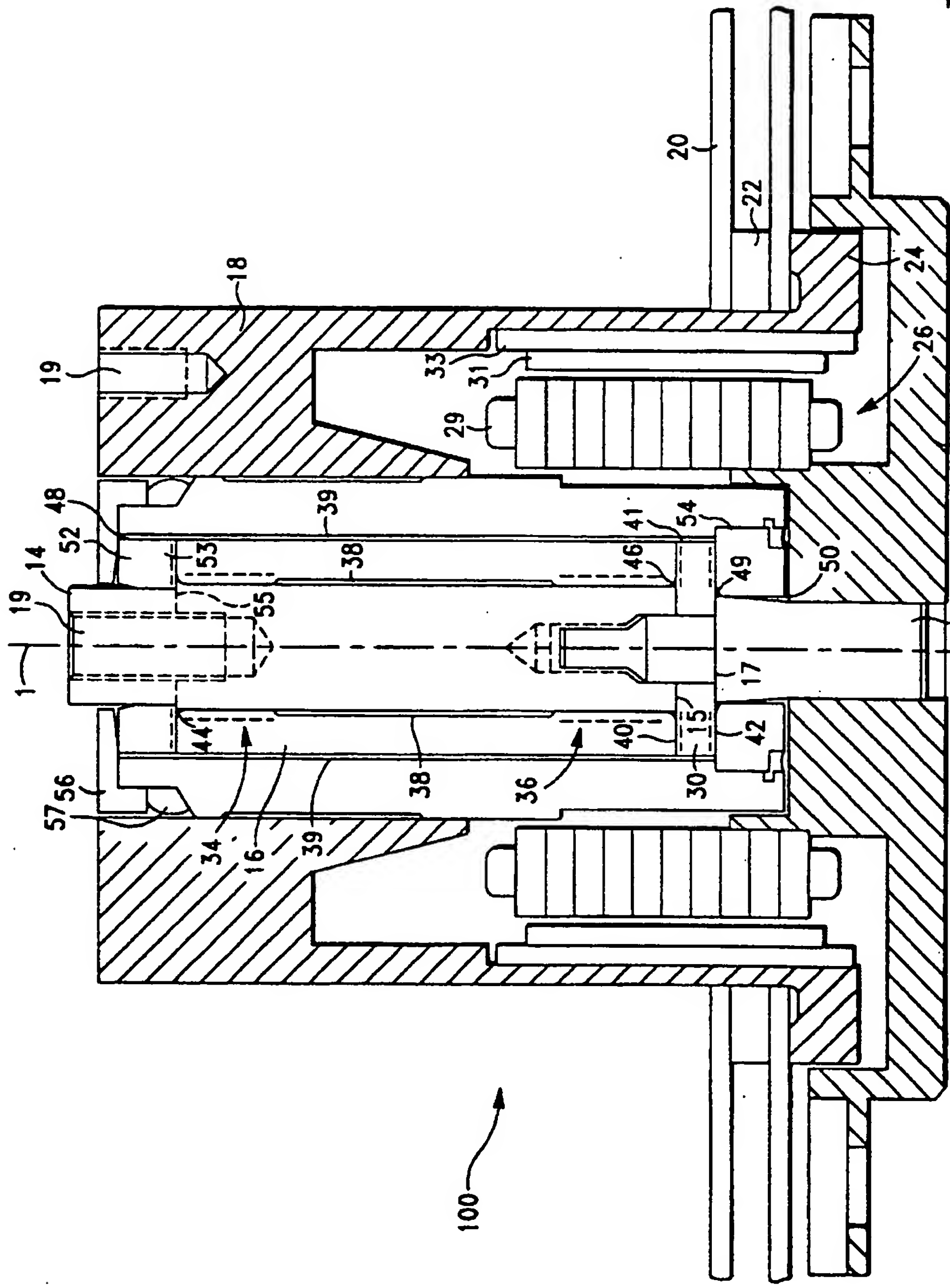

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FIG. -2

FIG. —3

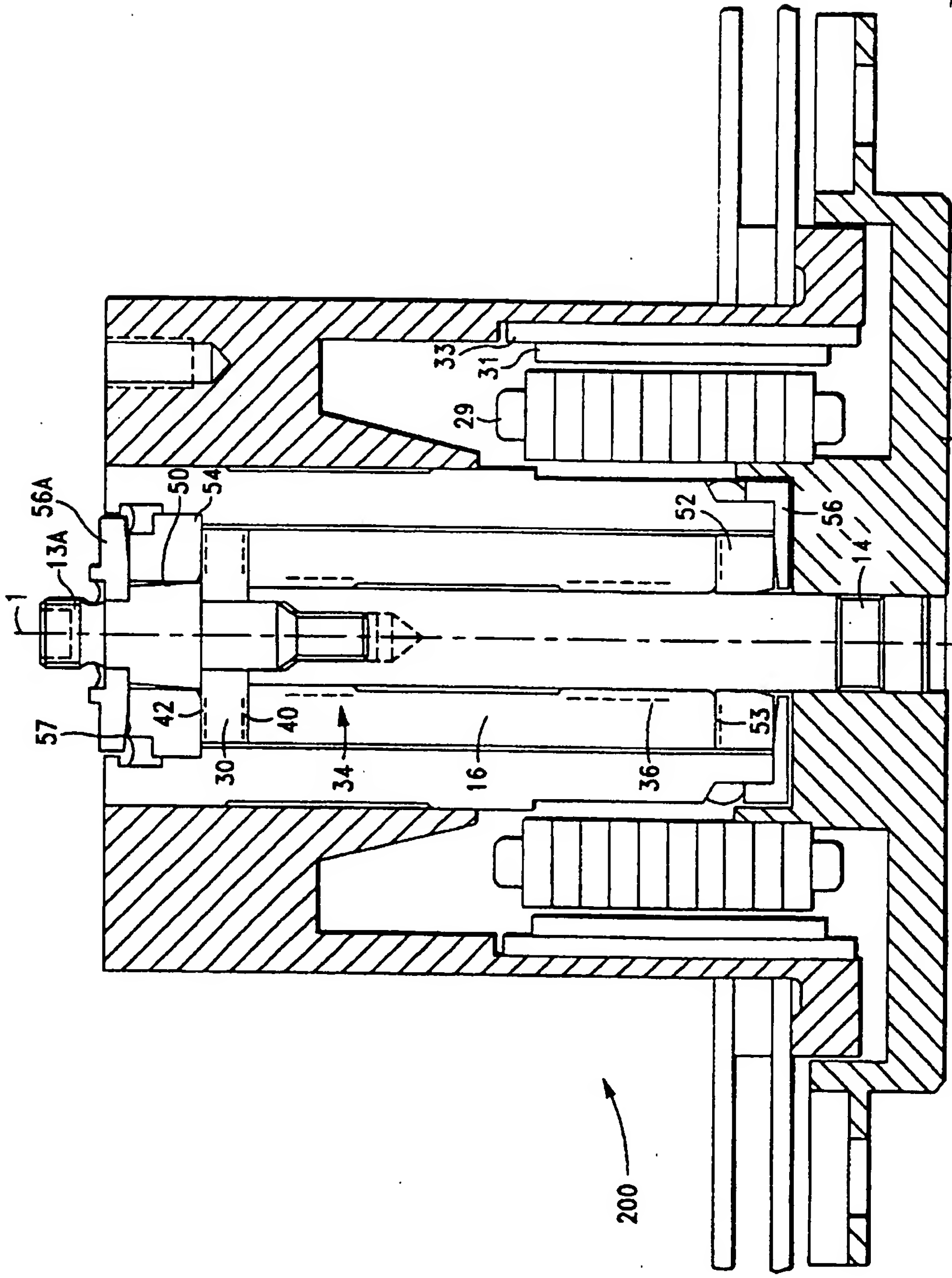
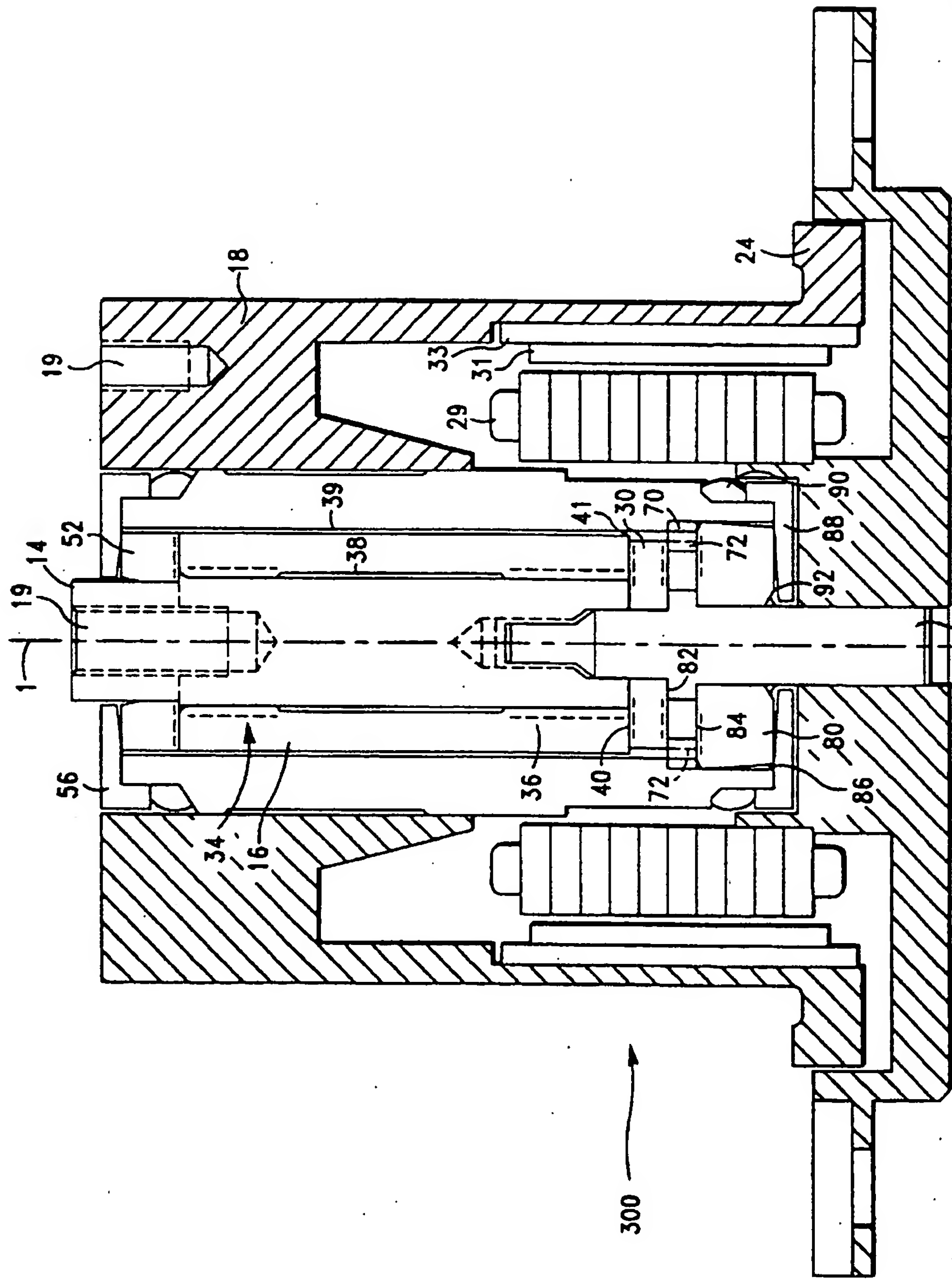


FIG. —4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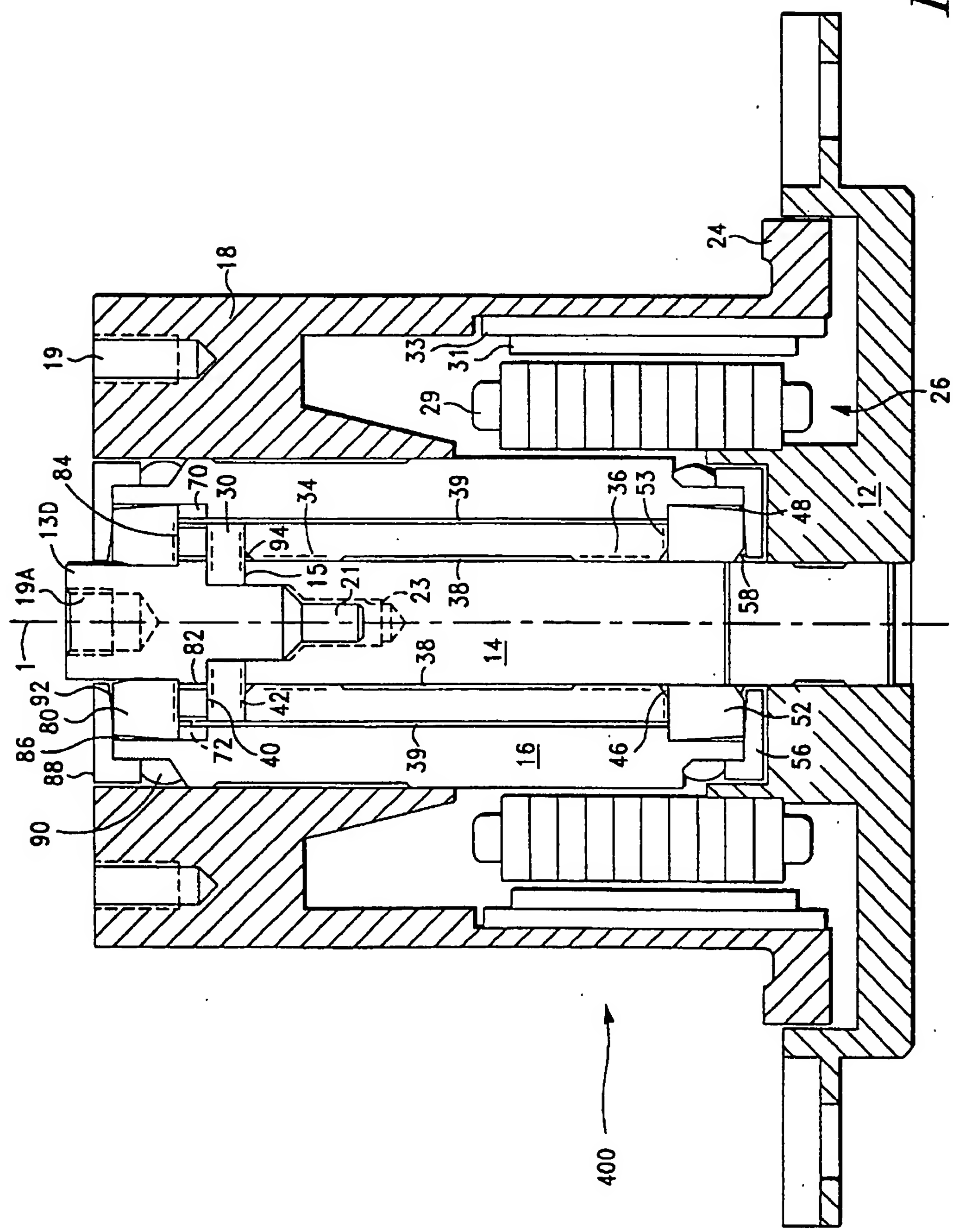
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FIG. — 5

HYDRODYNAMIC BEARING FOR SPINDLE MOTOR HAVING HIGH INERTIAL LOAD

FIELD OF THE INVENTION

The present invention relates to fluid bearings. More particularly, the present invention relates to a self-contained hydrodynamic beating unit for a spindle motor having a high inertial load and which includes a plurality of lubricating liquid seals for preventing escape of lubricating liquid.

BACKGROUND OF THE INVENTION

The development of computer hard disk drives demands ever increasingly higher track density, lower acoustic noise, and better reliability under shock and vibrational disturbances. The undesirable characteristics of the currently used ball bearing spindles, such as high non-repetitive runout, large acoustic noise, and high resonance frequencies due to bearing defect, impose severe limitation on the drive's capacity and performance.

The use of a non-contact bearing, such as a hydrodynamic bearing ("HDB"), may overcome the aforementioned limitation. The full film lubrication of a fluid bearing displays significantly lower non-repetitive runout and acoustic noise, and its higher damping provides better resistance to external shock and vibration. One example of a disk drive spindle motor including a HDB and centrifugal-capillary seals is found in the present inventors' (with another co-inventor) U.S. Pat. No. 5,423,612 entitled: "Hydrodynamic Bearing and Seal", the disclosure thereof being incorporated herein by reference.

The deployment of the HDB system in a hard disk drive environment requires that the lubricant be securely sealed inside of the bearing structure under all operating and non-operating conditions in order to prevent performance degradation of the bearing and contamination in the drive. At the same time, the bearing system needs to be easily manufacturable in order to satisfy cost requirements. As explained below these requirements often come into conflict with each other and have heretofore resulted in compromised HDB spindle designs.

There have been a number of prior approaches for providing seals for hydrodynamic bearing units. Static seals, such as O-rings, and dynamic clearance seals, such as surface tension or capillary seals, have been employed to seal hydrodynamic bearings.

One prior example is found in Hendler et al. U.S. Pat. No. 3,778,123 entitled: "Liquid Bearing Unit and Seal". In the Hendler et al. approach, a non-wettable liquid, such as mercury, is placed in an annular Vee-groove at an outside boundary of the hydrodynamic bearing system. In addition, a thin film of low vapor pressure vacuum pump oil is provided at an annular gap or space at the end of a journal member in order to retain the mercury seal. A pair of thin barrier films are also provided at the outer edge of the annular space to prevent the oil from spreading as a result of surface effects and/or centrifugal forces generated by relative rotation of the bearing system.

Another prior approach is found in Van Roemburg U.S. Pat. No. 4,596,474, entitled: "Bearing System Comprising Two Facing Hydrodynamic Bearings". In the Van Roemburg approach, two radial fluid bearings were separated by a central reservoir. Each bearing included a herringbone pattern, and the herringbone patterns were such that the outer legs of the Vee-grooves forming the herringbone pattern were longer than the inner legs. However, the system

maintained balanced pressure. This arrangement built up a lubricating liquid pressure at the apex of each Vee-groove which was greater than a counter pressure built up by the inner legs and by helical feed grooves which feed lubricant from a central reservoir area. By providing this differential pressure arrangement it is said that the lubricant was not pumped out of the bearing system.

A further prior approach is described in Anderson et al. U.S. Pat. No. 4,726,693, entitled: "Precision Hydrodynamic Bearing". The Anderson et al. approach uses a plurality of seals formed along the bearing unit including spiral grooves as well as an upper surface tension or capillary seal and a lower surface tension or capillary seal. However, the very nature of the Anderson et al. approach suggested that it was not adapted to omnidirectional operation or resistance to shock or vibratory forces.

Another prior approach is described in Titcomb et al. U.S. Pat. No. 4,795,275 and divisional patents U.S. Pat. Nos. 5,067,528 and 5,112,142, entitled: "Hydrodynamic Bearing". In the prior approaches described in these patents, surface tension dynamic seals were provided between axially extending surfaces of a thrust plate and bearing sleeve (or between tapered bearing surfaces). Pressure equalization ports were required and extended between the dynamic seals and interior lubricant reservoirs (or interior dynamic seals) to balance the hydrodynamic pressures in the lubricant in order to prevent the lubricant from being pumped through one of the dynamic seals. A method for introducing lubricating liquid into the hydrodynamic bearing employing a vacuum chamber and ultrasound is also described.

A similar prior approach is described in Pan U.S. Pat. No. 5,246,294 entitled: "Flow-Regulating Hydrodynamic Bearing". In this approach a disk spindle employs oppositely facing conical hydrodynamic bearing surfaces and a series of chambers and passages and a gravitational valve are provided to permit pressure-equalized centrifugally pumped global circulation of lubricating liquid drawn from one or more large reservoir volumes. A leak-preventing capillary trap "of minimum continuous axial length" may be provided at a clearance seal for passive capture of wandering lubricant when the bearing unit is at rest.

A further solution has been proposed by the present inventors with another in U.S. Pat. No. 5,423,612 entitled: "Hydrodynamic Bearing and Seal", the disclosure thereof being incorporated herein by reference. One drawback of the approach described in the '612 patent is it has proven somewhat difficult to provide recirculation ports around the bearings in order to realize a lubricant recirculation capability in circumstances such as imbalanced pumping and/or shock load. Another drawback was that since both top and bottom seals are at the inside diameter of the HDB unit, any splashed droplets which separate from the lubricant surface may be driven out of the bearing by centrifugal force. In addition, because of the small seal volume available at the HDB unit inside diameter, the lubricant may leak out of the bearing on account of thermal expansion and/or filling volume variations. A further drawback was that press fitting the thrust plate onto the shaft may cause excessive deformation resulting in large variations in bearing clearances and unacceptable hydrodynamic operation.

Small (3.5 inch disk diameter and smaller) form factor disk drives are used in unlimited applications and orientations. Consequently, a hydrodynamic bearing system for a disk spindle in such drives, e.g. having a full Z-dimension 1.6 inch height spindle manifesting high inertial loading, must also operate in all possible orientations, and to be able

to withstand and sustain certain shock events and vibration levels without leakage. A cover-secured or top-fixed HDB motor is required for disk drives with high inertial load, such as disk drives including six or more rotating disks. For top-fixed spindles, the requirement for two lubricant seals poses a considerable challenge.

Generally, there are two types of top-fixed HDB spindle designs, namely: single thrust-plate design, as illustrated in commonly assigned U.S. Pat. No. 5,423,612; and, double thrust-plate design, as illustrated in FIG. 1.

The single thrust-plate design of the type illustrated in U.S. Pat. No. 5,423,612 has the several drawbacks already noted above.

The double thrust-plate design is shown in FIG. 1, and it overcomes the first and second drawbacks of the conventional single thrust plate designs noted above, while the third drawback remains present. Unfortunately, there are some additional drawbacks. As shown in FIG. 1 a prior double thrust-plate hydrodynamic spindle bearing system **10** for a high performance miniature hard disk drive includes a base **12** and a shaft **14** fixed securely to the base **12** in a suitably sized opening **13** defined in the base **12**. A shaft housing **16** fits closely over the shaft **14** and cooperatively defines two hydrodynamic journal bearings **34** and **36**. A spindle hub **18** is attached to the shaft housing **16** and a flange **24** of hub **18** supports one or more data storage disks **20** (and spacers **22**) in a top-fixed arrangement(see e.g. FIG. 2). An in-hub spindle motor rotates the hub **18** and disks **20** relative to the base **12** and shaft **14** at a predetermined angular velocity.

An upper annular thrust bearing plate **28** fits securely over the shaft **14**, while a lower annular thrust bearing ring **30** also fits securely over the shaft **14**. Together, the plate **28** and ring **30** cooperate with adjacently facing radial faces of the shaft housing **16** to provide an upper hydrodynamic axial thrust bearing **40** and a lower hydrodynamic axial thrust bearing **42**. A central axial reservoir region **38** is provided for lubricating liquid between the two radial hydrodynamic bearings **34** and **36**. Two end reservoirs **44** and **46** are formed respectively between the bearings **34** and **40**, and the bearings **36** and **42**.

Two surface tension annular capillary seals **48** and **50** are provided in annular gaps outwardly beyond the two thrust bearings **40** and **42** relative to the shaft **14**. The upper seal **48** is formed by outwardly axially divergent, oppositely facing cylindrical walls of the thrust plate **28** and shaft housing **16**, and the lower seal **50** is formed by outwardly axially divergent, oppositely facing cylindrical walls of the thrust ring **30** and the shaft housing **16**. In these seals **48** and **50**, a curved lubricant-air interface typical of a surface tension interface is located approximately midway of the gap.

Two oil containment bushings **52** and **54** are secured in a sealed arrangement to the shaft housing **16** as shown in FIG. 1. The seal **48** is shown as having a radially inward extension **49**, which effectively extends the upper capillary seal. The lower capillary seal **50** is likewise extended radially inwardly by an extension **51**. Further details of this prior arrangement are disclosed in commonly assigned, copending U.S. patent application Ser. No. 08/363,566 filed on Dec. 22, 1994, entitled: "A Self-Contained Hydrodynamic Bearing Unit" now U.S. Pat. No. 5,558,445, the disclosure thereof being incorporated herein by reference.

As noted above, there are several additional drawbacks with the double thrust-plate design illustrated in FIG. 1. One additional drawback is controlling the tolerance of the total length of the sleeve which defines the thrust bearing clear-

ance (which is about 10 microns). A second drawback relates to manufacturing difficulty in controlling the tolerances of perpendicularity and surface finish at both ends of the e.g. bronze sleeve. Third, because the sleeve is typically of bronze, the sleeve tends to wear by coming into contact with the grooved steel thrust plate having pumping grooves during starting and stopping intervals. Fourth, it has proven difficult to apply adhesive to seal the thrust plate/shaft press-fit areas of the bearing unit. Adhesive grooves at the side of the bottom thrust plate cause a lack of symmetry in the thrust plate and additional deformation during press fitting of the thrust plate and shaft.

Thus, a hitherto unsolved need has remained for a hydrodynamic bearing system having a high inertial load which is leak free irrespective of orientation, shock and vibration, and which is readily and reliably manufacturable at reasonably low manufacturing cost.

SUMMARY OF THE INVENTION WITH OBJECTS

A general object of the present invention is to provide a self-contained hydrodynamic bearing system for a spindle manifesting high inertial loading which overcomes limitations and drawbacks of the prior art.

Another object of the present invention is to provide a hydrodynamic bearing system which minimizes the risk of lubricating liquid leakage under all required operating conditions e.g. for disk drive spindle applications.

Yet another object of the present invention is to provide a hydrodynamic bearing system which combines improved capillary seals while providing for internal lubricating liquid recirculation.

One more object of the present invention is to provide a self-contained hydrodynamic bearing system for a spindle manifesting high inertial loading which is readily manufacturable at low cost.

Another object of the present invention is to provide a simplified hydrodynamic bearing system which includes upper and lower capillary seals for trapping and containing lubricating liquid at annular seal locations axially outside of journal bearing locations, and radially outward of a shaft-sleeve diameter of the system, an arrangement for returning liquid trapped at the secondary seal to the primary seal, and an arrangement for recirculating lubricating liquid between upper and lower seals.

A further object of the present invention is to provide a hydrodynamic bearing design for a disk drive spindle assembly which virtually eliminates lubricating liquid leakage thereby to improve substantially the useful life of the bearing and disk drive.

One more object of the present invention is to provide a hydrodynamic bearing assembly for a disk drive spindle and including both radial hydrodynamic bearings and axial thrust hydrodynamic bearings and a plurality of seals in an arrangement facilitating manufacture and assembly and leading to virtual leak-free operation, even when subjected to shock or vibration energy.

Still one more object of the present invention is to provide a simplified hydrodynamic bearing design which is simpler to manufacture, which operates reliably in any angular orientation, and which achieves superior bearing longevity over prior art fluid bearing designs.

In accordance with principles of the present invention, a self-contained hydrodynamic bearing unit comprises a shaft defining a threaded axial opening at one end thereof and a

shaft shoulder adjacent the opening and perpendicular with a longitudinal axis of the shaft, a sleeve defining an opening for receiving the shaft for relative rotation, a pair of longitudinally spaced-apart radial hydrodynamic journal bearings defined between the shaft and the sleeve, a shaft-bolt including a threaded end region for mating with the threaded axial opening of the shaft and defining a bolt shoulder, an annular thrust plate having two parallel radial faces and adapted to fit upon the shaft-bolt for mounting between the shaft shoulder and the shaft bolt shoulder when the shaft-bolt is mated with the threaded axial opening of the shaft, the sleeve defining a radial thrust bearing surface portion extending radially outwardly at one end and positioned to confront one radial face of the thrust plate, an annular thrust bushing mounted to the sleeve and defining a radial thrust bearing surface portion confronting the other radial face of the thrust plate, the annular thrust plate defining two axial hydrodynamic thrust bearings respectively with the radial thrust bearing surface portion of the sleeve and the radial thrust bearing surface portion of the annular thrust bushing and a hydrodynamic thrust bearing lubricant reservoir between the two axial hydrodynamic thrust bearings and between a cylindrical face of the thrust plate and a facing cylindrical wall of the sleeve, and a hydrodynamic bearing lubricant in the bearing unit and at the pair of hydrodynamic journal bearings and at the two axial hydrodynamic thrust bearings and in the hydrodynamic bearing lubricant reservoir.

Capillary seals at diameters outside of the diameter of the shaft-sleeve radial journal bearings, and hydrodynamic lubricant recirculation paths for the hydrodynamic radial bearings and the two hydrodynamic thrust bearings are also aspects of the present invention.

These and other objects, advantages, aspects and features of the present invention will be more fully understood and appreciated by those skilled in the art upon consideration of the following detailed description of a preferred embodiment, presented in conjunction with the accompanying drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

In the Drawings:

FIG. 1 is an enlarged diagrammatic view in section and elevation of a right one half side of a prior self contained hydrodynamic bearing unit and seals within a disk drive spindle.

FIG. 2 is an enlarged view in elevation and section of a single thrust, single seal plate, thrust down HDB spindle unit in accordance with principles of the present invention.

FIG. 3 is an enlarged view in elevation and section of a single thrust, single seal plate, thrust up HDB spindle unit in accordance with principles of the present invention.

FIG. 4 is an enlarged view in elevation and section of a single thrust plate down, double seal plate HDB spindle unit in accordance with the principles of the present invention.

FIG. 5 is an enlarged view in elevation and section of a single thrust plate up, double seal plate HDB spindle unit in accordance with principles of the present invention.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

Turning to FIG. 2, a single thrust, single seal plate, thrust down HDB spindle unit **100** incorporating principles of the present invention includes a base **12**, and a lower shaft-bolt **13** having a threaded stud portion **21** for mating with a threaded opening **23** of a central shaft **14** for securing the central shaft **14** to the base. The shaft **14** is concentric about a longitudinal axis **1** of symmetry-rotation. The shaft **14** defines a threaded axial opening **19** at the top for receiving a screw (not shown) for securing the shaft **14** to a top cover to provide structural rigidity to the spindle unit **100**. A bronze sleeve **16** fits over the shaft with a close clearance tolerance and defines two axially spaced apart hydrodynamic journal bearings **34** and **36** by virtue of pumping grooves suitably defined in the sleeve (or shaft) at the beating regions. A spindle hub **18** is secured to the sleeve **16**. An upper radial wall of the hub **18** defines a series of threaded bores **19** for receiving screws for securing a clamp (not shown) for securely clamping a stack of disks **20** and spacers **22** to a lower flange **24** of the hub. A DC brushless in-hub spindle motor **26** includes a fixed stator assembly **29** with slots facing alternating magnet pole faces of a permanent magnet **31** secured to an annular flux return ring **33**, in turn secured to an inside cylindrical wall of the hub **18** in conventional fashion. Driving currents selectively applied to various coils of the stator assembly induce reaction forces in the permanent magnet, and rotation of the sleeve **16** and hub **18** ensues. As thus far described, the spindle unit **100** is a top-fixed or cantilevered spindle design manifesting high inertial loading.

A "thrust-down" annular thrust plate **30** e.g. of steel alloy is guided onto the lower shaft-bolt **13** in a close-tolerance slip-fit arrangement. A lower radial wall **15** of the shaft **14** establishes perpendicularity of the thrust plate **30** relative to the axis **1**, and a shoulder **17** formed as a base of a conical portion of the shaft-bolt **13** applies a securing force to secure the thrust plate **30** in place against the wall **15** when the shaft-bolt **13** is tightened. An e.g. bronze thrust bushing **54** is secured to an inner annular recess of the sleeve **16**, and sandwiches the thrust plate **30** in a precision hydrodynamic bearing relationship between a lower radial face of the sleeve **16** and an upper radial face of the thrust plate **30**. Pumping grooves defined on the thrust plate **30** or on the upper and lower radial faces of the sleeve **16** and thrust bushing **54** respectively define upper and lower hydrodynamic axial thrust bearings **40** and **42** with typical nominal clearances of approximately 10 microns.

An upper seal plate **52** is press-fit onto the shaft **14** until a shoulder **55** is reached which aligns the seal plate **52** precisely with regard to an upper radial wall of the sleeve **16**. The seal plate **52** functions to provide an upper radial seal for lubricant and extend the seal to a location radially outwardly of the inside diameter of the shaft-sleeve journal diameter in order to reduce loss of lubricant otherwise resulting from centrifugal force during operation of the bearing unit **100**. Since the seal plate **52** functions merely as a seal, and not as an upper thrust bearing as shown in the FIG. 1 example, a much greater clearance, such as 50–100 microns, may be provided between the seal plate **52** and the upper radial face of the sleeve **16**. Sealing grooves **53** may be defined in one or both of the opposite faces of the seal plate **52** and the upper radial face of the sleeve **16** to oppose lubricant displacement otherwise induced by centrifugal force during bearing unit operation. The seal plate **52** forms an upper annular capillary seal **48** which is radially outside of the shaft-sleeve interface defining the journal bearings **34** and **36** to reduce lubricant losses from centrifugal forces as noted above. A containment ring **56** forms an upper, secondary lubricant containment capillary **58** with an upper face of the seal plate **52** which may extend the upper capillary seal **48** or may exist in addition thereto. The containment ring **56** is secured to an annular end portion of

the sleeve 16 by a suitable adhesive 57. Facing surfaces of the containment ring 56 and the seal plate 52 may be coated with a suitable thin film barrier material for impeding surface flow of the lubricating liquid droplets out of the containment capillary 58.

As shown, the top-fixed spindle unit 100 provides lubricant recirculation paths between the journal bearings 34 and 36 by virtue of an inside annular passageway 38 next to the shaft 14, and one or more radially outward longitudinal bores 39 which are in substantial radial alignment with the upper annular capillary seal 48 and a reservoir 41 just beyond an outer cylindrical wall of the thrust plate 30. The reservoir 41 also feeds and buffers lubricant to and from the HDB thrust bearings 40 and 42. Upper and lower chamfers defined at inside corners of the sleeve 16 adjacent to the shaft 14 provide additional storage reservoirs 44 and 46 for the hydrodynamic fluid lubricant. A lower annular reservoir 49 is positioned between the lower thrust bearing 42 and a lower annular capillary seal 50. The narrow (e.g ½ millimeter diameter longitudinal bores through the sleeve 16 are preferably defined by wire electro-discharge machining (EDM) techniques.

FIG. 3 presents an alternative spindle unit 200 in which the thrust plate 30 is secured to the top of the shaft 14 by an upper threaded shaft-bolt 13A which includes a threaded boss for receiving a nut securing a cover onto the shaft shaft-bolt 13A and in turn securing the shaft 14. In FIG. 3, the elements of the FIG. 2 embodiment associated with the thrust plate have been moved to the top of the unit 200, while the elements of the FIG. 2 embodiment at the top of the shaft 14, have been displaced to the bottom, without changing the reference numerals for these elements between the two figures. In FIG. 3, the threaded shaft-bolt 13A also secures an upper containment ring 56A. A secondary lubricant containment seal 57 is formed between the upper containment ring 56A and the upper thrust bushing 54 in order to reduce the likelihood of lubricant leakage from the top seal. Otherwise, the units 100 and 200 are structurally and functionally equivalent.

While overcoming many of the problems of the prior single and double thrust plate designs discussed in the background of the invention, there remain two drawbacks with the FIGS. 2 and 3 embodiments. First, there are no lubricant recirculation ports around one of the two thrust bearings for facilitating lubricant recirculation and pressure equilibrium. Second, one capillary seal 50 is at the bearing unit inside diameter (i.e. shaft-sleeve HDB journal bearing interface).

These remaining drawbacks are overcome within the embodiments of FIGS. 4 and 5. In FIG. 4 a single down thrust plate, double seal HDB spindle unit 300 follows in many important respects the spindle unit 100 discussed above in conjunction with FIG. 2. There are several important differences with regard to the thrust plate 30. In the FIG. 4 spindle unit 300 the single thrust plate 30 is fixed onto the shaft 14 by the threaded shaft-bolt 13B. The thrust plate 30 provides HDB thrust bearings 40 and 42, except that in this embodiment 300, a thrust bushing 70 is provided which is secured e.g. by press-fitting into an inner cylindrical recess of the sleeve 16 and rotates with it. The lower thrust bearing 42 is formed at a top radial wall of the thrust bushing 70 and the lower radial wall of the thrust plate 30. One or several lubricant recirculation ports 72 are defined through the thrust bushing 70 to enable recirculation of hydrodynamic lubricating liquid. A bottom seal plate 80 is also press-fit onto the threaded shaft-bolt 13B. The shaft-bolt 13B defines an annular boss 82 which enables precise positioning of the thrust plate 30 and bottom seal plate 80 relative to the thrust bushing 70. Pumping grooves 84 may be defined between the lower radial face of the thrust bushing 70 and the bottom seal plate 80 to neutralize radial pumping action otherwise attributable to relative rotation and centrifugal forces. A radially outer annular capillary seal 86 is defined between an inside wall of the sleeve 16 and an outer tapered face of the bottom seal plate 80. A bottom lubricant containment ring 88 includes an upturned flange which is secured to an outer cylindrical region of the sleeve 16 by a suitable adhesive 90. The containment ring 88 cooperates with a bottom radial wall of the bottom seal plate 80 to define a secondary capillary seal-lubricant trap 92.

FIG. 5 presents an alternative HDB spindle unit 400 which modifies the FIG. 4 embodiment 300 by placing the single thrust plate 30 at the top of the shaft 14. A shaft-bolt 13D provides a widened threaded opening 19A enabling the cover (not shown) to be secured to the shaft-bolt 13D without disturbing its relative alignment with the shaft 14, and sleeve 16. Otherwise, the description given above for the FIG. 4 embodiment 300 applies to the embodiment 400 of FIG. 5.

The embodiments 300 and 400 of FIGS. 4 and 5 respectively overcome essentially all of the problems mentioned above for the prior designs, and improve somewhat upon the embodiments of FIGS. 2 and 3. The thrust bearing clearance is controlled by the thickness of the thrust plate 30 and the height between the facing radial shoulder of the sleeve 16 and the step of the sleeve above the thrust plate for receiving the thrust bushing 70, whose dimensional tolerances are easier to control in manufacturing. Selective assembly may also be employed to control further the thrust bearing clearances. Press fitting of the thrust bearing 30 relative to the sleeve 16 is avoided in these FIGS. 4 and 5 embodiments 300 and 400. Recirculation ports 39 and 72 facilitate recirculation of hydrodynamic lubricant around the journal bearings 34 and 36, and around the thrust bearings 40 and 42. Both the outer axial capillary seals 48 and 86 are at an outer diameter of the bearing unit 300, 400 and are generally aligned with the passages 39 and 72.

Furthermore, the FIG. 5 embodiment 400 further reduces wear of the sleeve 16 if the thrust bushing 70 is made of stainless steel and the spindle unit 400 is mounted for the most part in the up-down orientation given it in the FIG. 5 view. Tolerance of the total length of the sleeve 16 is relaxed because of relatively large allowable clearance (e.g. 50–100 microns) between the seal plate 52 and the sleeve 16. Only the radial face of the sleeve 16 directly facing the thrust plate 30 needs to be provided with a tightly controlled surface finish and maintain close perpendicularity tolerances. Thus, the embodiments 300 and 400 of FIGS. 4 and 5 maintain all of the advantageous lubricant seal arrangements found in the prior double-thrust plate design 10 described in connection with FIG. 1 without having the same drawbacks and manufacturing difficulties.

As used herein, the expressions indicating orientation such as "upper", "lower", "top", "bottom", "height" and the like, are applied in a sense related to normal viewing of the figures rather than in any sense of orientation during particular operation, etc. These orientation labels are provided simply to facilitate and aid understanding of the figures and should not be construed as limiting.

To those skilled in the art, many changes and modifications will be readily apparent from consideration of the foregoing description of a preferred embodiment without departure from the spirit of the present invention, the scope

thereof being more particularly pointed out by the following claims. The descriptions herein and the disclosures hereof are by way of illustration only and should not be construed as limiting the scope of the present invention which is more particularly pointed out by the following claims.

What is claimed is:

1. A self-contained hydrodynamic bearing unit comprising:

a shaft defining a threaded axial opening at one end thereof and a shaft shoulder adjacent the opening and perpendicular with a longitudinal axis of the shaft,

a sleeve defining an opening for receiving the shaft for relative rotation,

a pair of longitudinally spaced-apart radial hydrodynamic journal bearings defined between the shaft and the sleeve,

a shaft-bolt including a threaded end region for mating with the threaded axial opening of the shaft and defining a bolt shoulder,

an annular thrust plate having two parallel radial faces and adapted to fit upon the shaft-bolt for mounting between the shaft shoulder and the shaft bolt shoulder when the shaft-bolt is mated with the threaded axial opening of the shaft,

the sleeve defining a radial thrust bearing surface portion extending radially outwardly at one end and positioned to confront one radial face of the thrust plate,

an annular thrust bushing mounted to the sleeve and defining a radial thrust bearing surface portion confronting the other radial face of the thrust plate,

the annular thrust plate defining two axial hydrodynamic thrust bearings respectively with the radial thrust bearing surface portion of the sleeve and the radial thrust bearing surface portion of the annular thrust bushing and a hydrodynamic thrust bearing lubricant reservoir between the two axial hydrodynamic thrust bearings and between a cylindrical face of the thrust plate and a facing cylindrical wall of the sleeve,

hydrodynamic bearing lubricant in the bearing unit and at the pair of hydrodynamic journal bearings and at the two axial hydrodynamic thrust bearings and in the hydrodynamic bearing lubricant reservoir.

2. The self-contained hydrodynamic bearing unit set forth in claim 1 further comprising an annular seal plate secured to the shaft at an end region of the shaft opposite an end mounting the thrust plate, the annular seal plate defining an annular capillary seal for containing the hydrodynamic bearing lubricant at a diameter greater than a diameter of the radial journal bearings defined between the shaft and the sleeve.

3. The self-contained hydrodynamic bearing unit set forth in claim 2 wherein an inner cylindrical wall defined by the sleeve and adjacently facing surface of the annular seal plate define a divergent gap for the annular capillary seal.

4. The self-contained hydrodynamic bearing unit set forth in claim 2 wherein the annular seal plate defines a radial face in confronting relationship with a radial surface portion of the sleeve, and further comprising pumping grooves defined in at least one of the radial face of the annular seal plate and the radial surface portion of the sleeve for pumping the hydrodynamic bearing lubricant toward the shaft to resist centrifugal forces otherwise pumping the hydrodynamic bearing lubricant radially outwardly during relative rotation of the bearing unit.

5. The self-contained hydrodynamic bearing unit set forth in claim 2 further comprising an annular containment ring having a cylindrical outer flange secured to an annular ring of the sleeve and surrounding the annular seal plate, defining a secondary hydrodynamic bearing lubricant containment seal in communication with the annular capillary seal for trapping lubricant otherwise escaping from the annular capillary seal.

6. The self-contained hydrodynamic bearing unit set forth in claim 5 wherein the secondary hydrodynamic beating lubricant containment seal has surfaces coated with a thin film barrier material for impeding surface flow of droplets of the lubricant.

7. The self-contained hydrodynamic bearing unit set forth in claim 2 further comprising a hydrodynamic bearing lubricant reservoir in an annulus defined between the shaft and sleeve between the pair of radial journal bearings and at least one longitudinal recirculation hole defined in the shaft at a diameter approximately the diameter of the annular capillary seal and extending through the sleeve to a hydrodynamic bearing lubricant reservoir adjacent the thrust plate.

8. The self-contained hydrodynamic bearing unit set forth in claim 1 wherein the annular thrust bushing and the shaft bolt define a divergent gap annular capillary seal for the hydrodynamic bearing lubricant at a diameter approximately at the diameter of the radial journal bearings defined between the shaft and the sleeve.

9. The self-contained hydrodynamic bearing unit set forth in claim 1 further comprising a hydrodynamic bearing lubricant reservoir in an annulus defined between the shaft and sleeve between the pair of radial journal bearings.

10. The self-contained hydrodynamic bearing unit set forth in claim 1 further comprising:

an annular seal plate secured to the shaft at an end region of the shaft opposite an end mounting the thrust plate, the annular seal plate defining a first annular capillary seal for containing the hydrodynamic bearing lubricant at a diameter greater than a diameter of the radial journal bearings defined between the shaft and the sleeve,

wherein the annular thrust bushing and the shaft-bolt define a second divergent gap annular capillary seal for the hydrodynamic bearing lubricant at a diameter approximately at the diameter of the radial journal bearings defined between the shaft and the sleeve,

a hydrodynamic bearing lubricant reservoir in an annulus defined between the shaft and sleeve between the pair of radial journal bearings, and

at least one longitudinal recirculation hole defined in the shaft at a diameter approximately the diameter of the annular capillary seal and extending through the sleeve to a hydrodynamic bearing lubricant reservoir adjacent the thrust plate.

11. The self-contained hydrodynamic bearing unit set forth in claim 1 further comprising an annular seal plate secured to the shaft-bolt adjacent to the annular thrust bushing, the annular seal plate and an adjacent facing wall segment of the sleeve defining an annular seal at a diameter greater than a diameter of the radial journal bearings defined between the shaft and the sleeve.

12. The self-contained hydrodynamic bearing unit set forth in claim 11 wherein the annular seal plate and the adjacent facing wall segment of the sleeve define the annular seal as a divergent gap annular capillary seal.

13. The self-contained hydrodynamic bearing unit set forth in claim 11 wherein the annular thrust bushing defines at least one hydrodynamic bearing lubricant recirculation port in communication with the hydrodynamic thrust bearing lubricant reservoir for enabling recirculation of hydro-

dynamic bearing lubricant at the thrust bearings during operation of the bearing unit.

14. The self-contained hydrodynamic bearing unit set forth in claim 1 further comprising a base and a cover, the shaft-bolt being secured to the base, and the shaft being secured to the cover.

15. The self-contained hydrodynamic bearing unit set forth in claim 1 further comprising a base and a cover, the shaft-bolt being secured to the cover, and the shaft being secured to the base.

16. A self-contained hydrodynamic bearing unit comprising:

a shaft defining a threaded axial opening at one end thereof and a shaft shoulder adjacent the opening and perpendicular with a longitudinal axis of the shaft,

a sleeve defining an opening for receiving the shaft for relative rotation,

a pair of longitudinally spaced-apart radial hydrodynamic journal bearings defined between the shaft and the sleeve,

a shaft-bolt including a threaded end region for mating with the threaded axial opening of the shaft and defining a bolt shoulder,

an annular thrust plate having two parallel radial faces and adapted to fit upon the shaft-bolt for mounting between the shaft shoulder and the shaft bolt shoulder when the shaft-bolt is mated with the threaded axial opening of the shaft,

the sleeve defining a radial thrust bearing surface portion extending radially outwardly at one end and positioned to confront one radial face of the thrust plate,

an annular thrust bushing mounted to the sleeve and defining a radial thrust bearing surface portion confronting the other radial face of the thrust plate,

the annular thrust plate defining two axial hydrodynamic thrust bearings respectively with the radial thrust bearing surface portion of the sleeve and the radial thrust bearing surface portion of the annular thrust bushing and a hydrodynamic thrust bearing lubricant reservoir between the two axial hydrodynamic thrust bearings and between a cylindrical face of the thrust plate and a facing cylindrical wall of the sleeve,

a first annular seal plate secured to the shaft-bolt adjacent to the annular thrust bushing, the annular seal plate and an adjacent facing wall segment of the sleeve defining a first annular seal at a diameter greater than a diameter of the radial journal beatings defined between the shaft and the sleeve, and

hydrodynamic bearing lubricant in the bearing unit and at the pair of hydrodynamic journal bearings and at the two axial hydrodynamic thrust bearings and in the hydrodynamic bearing lubricant reservoir.

17. The self-contained hydrodynamic bearing unit set forth in claim 16 further comprising a second annular seal plate secured to the shaft at an end region of the shaft opposite an end mounting the thrust plate, the annular seal plate defining a second annular capillary seal for containing the hydrodynamic bearing lubricant at a diameter greater than a diameter of the radial journal bearings defined between the shaft and the sleeve.

18. The self-contained hydrodynamic bearing unit set forth in claim 16 wherein the annular thrust bushing defines at least one hydrodynamic bearing lubricant recirculation port in communication with the hydrodynamic thrust bearing lubricant reservoir for enabling recirculation of hydrodynamic bearing lubricant at the thrust bearings during operation of the beating unit.

19. The self-contained hydrodynamic bearing unit set forth in claim 18 further comprising a second annular seal plate secured to the shaft at an end region of the shaft opposite an end mounting the thrust plate, the annular seal plate defining a second annular capillary seal for containing the hydrodynamic bearing lubricant at a diameter greater than a diameter of the radial journal bearings defined between the shaft and the sleeve, and a hydrodynamic bearing lubricant reservoir in an annulus defined between the shaft and sleeve between the pair of radial journal bearings and at least one longitudinal recirculation hole defined in the shaft at a diameter approximately the diameter of the second annular capillary seal and extending through the sleeve to the hydrodynamic thrust bearing lubricant reservoir adjacent the thrust plate.

20. A self-contained hydrodynamic bearing unit comprising:

a base,

a shaft fixed at one end to the base, and providing a cover attachment means at another end for securing a cover fixed relative to the base,

a sleeve having a central opening for receiving the shaft in a rotational relationship,

at least one hydrodynamic journal bearing defined between confronting cylindrical walls of the shaft and sleeve,

a hydrodynamic lubricant reservoir defined between the shaft and sleeve for providing the journal bearing with a supply of lubricant,

a double-faced single thrust plate fixed to the shaft and located at one end of the sleeve,

a first hydrodynamic thrust bearing formed at confronting radial faces of the thrust plate and sleeve, the first hydrodynamic thrust bearing communicating with the hydrodynamic lubricant reservoir,

a first shaft seal structure having at least one part thereof secured to the shaft and forming a second hydrodynamic bearing at confronting radial faces of the one part and the thrust plate, the second hydrodynamic thrust bearing communicating with the hydrodynamic lubricant reservoir via the first hydrodynamic thrust bearing,

the first shaft seal structure forming a first axial capillary seal located between the first shaft seal structure and one of the sleeve and shaft and exteriorly of the second hydrodynamic thrust bearing for sealing hydrodynamic lubricant against escape during relative rotation between the shaft and the sleeve.

21. The self-contained hydrodynamic bearing unit set forth in claim 20 wherein the shaft includes a threaded axial opening and a shaft shoulder, and further comprising a shaft-bolt including a threaded end region for mating with the threaded axial opening and defining a shaft bolt shoulder, the annular thrust plate being mounted between the shaft shoulder and the shaft bolt shoulder.

22. The self-contained hydrodynamic bearing unit set forth in claim 20 wherein the thrust plate and first shaft seal structure are located at an end of the shaft adjacent to the cover attachment means.

23. The self-contained hydrodynamic bearing unit set forth in claim 20 wherein the first shaft seal structure comprises a thrust bushing secured to the sleeve and having a radial face confronting a radial face of the thrust plate thereby forming the second hydrodynamic thrust bearing,

and a first seal plate secured to the shaft exteriorly of the thrust plate and thrust bushing, the first seal plate forming the first axial capillary seal at a radius greater than a radius of the hydrodynamic journal bearing.

24. The self-contained hydrodynamic bearing unit set forth in claim 21 wherein the sleeve defines at least one sleeve hydrodynamic lubricant circulation passage axially aligned at an outer radius of the thrust plate for circulating lubricant between the hydrodynamic lubricant reservoir and the first hydrodynamic thrust bearing, and wherein the thrust bushing defines at least one bushing hydrodynamic lubricant circulation passage axially aligned with the sleeve hydrodynamic lubricant circulation passage, for communicating lubricant between the thrust bearing and the first axial capillary seal.

25. The self-contained hydrodynamic bearing unit set forth in claim 21 wherein the first shaft seal structure further includes a first lubricant containment ring secured to the sleeve exteriorly of the first seal plate.

26. The self-contained hydrodynamic bearing unit set forth in claim 25 wherein the first shaft seal structure further includes a radial capillary seal defined between the first lubricant containment ring and the first seal plate and communicating with the axial capillary seal.

27. The self-contained hydrodynamic bearing unit set forth in claim 20 wherein the sleeve defines at least one hydrodynamic lubricant recirculation passage axially aligned at an outer radius of the thrust plate for circulating lubricant between the hydrodynamic lubricant reservoir and the first hydrodynamic thrust bearing.

28. The self-contained hydrodynamic bearing unit set forth in claim 20 further comprising a second shaft seal structure including a second seal plate secured to the shaft at an end opposite to the end of the shaft having the first shaft seal structure, the second shaft seal structure forming a second axial capillary seal located between the second shaft seal structure and the sleeve at a radius greater than a radius of the hydrodynamic journal bearing for sealing hydrodynamic lubricant against escape during relative rotation between the shaft and the sleeve.

29. The self-contained hydrodynamic bearing unit set forth in claim 28 wherein the second shaft seal structure further includes a second lubricant containment ring secured to the sleeve exteriorly of the second seal plate.

30. The self-contained hydrodynamic bearing unit set forth in claim 29 wherein the second shaft seal structure further includes a radial capillary seal defined between the second lubricant containment ring and the second seal plate and communicating with the second axial capillary seal.

* * * * *

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United States Patent [19]

Oku

[11] **Patent Number: 6,097,121**

[45] **Date of Patent: Aug. 1, 2000**

[54] **SPINDLE MOTOR DESIGN FACILITATING AND ENSURING PRECISE ASSEMBLY OF THE MOTOR**

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[73] Assignee: **Nidec Corporation**, Kyoto, Japan

[21] Appl. No.: **09/110,089**

[22] Filed: **Jul. 2, 1998**

[51] **Int. Cl.**[7] **H02K 5/00**

[52] **U.S. Cl.** .... **310/91**; 310/42; 310/90; 310/67 R; 310/68 R; 310/258; 310/71

[58] **Field of Search** .... 310/42, 67 R, 310/68 R, 90; 360/98.07, 99.04, 99.07, 99.08; 384/107

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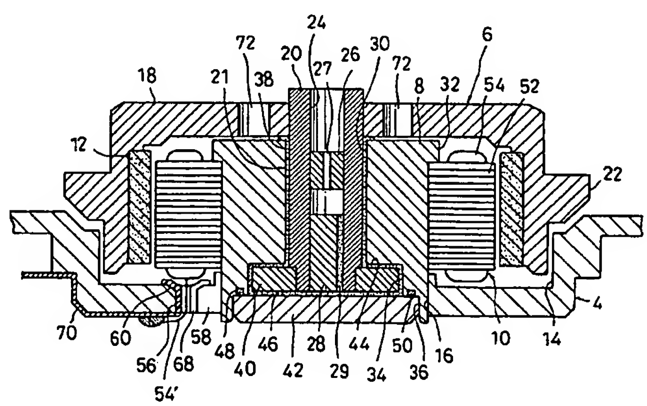
Primary Examiner—Elvin Enad

Attorney, Agent, or Firm—McCormick, Paulding & Huber LLP

[57] **ABSTRACT**

A spindle motor comprises a stationary member having a round mounting hole therein, a tubular member fitted into the mounting hole of the stationary member, a bearing means mounted to the inner side of the tubular member, a rotor supported by the bearing means for rotation in relation to the tubular member, a stator fitted onto the outer side of the tubular member, and a rotor magnet mounted to the rotor to be located radially and outwardly of the stator. The procedure of assembling the spindle motor includes mounting the rotor magnet to the rotor, joining the rotor by the bearing means to the tubular member for rotation, fitting the stator onto the tubular member, and fitting the tubular member into the mounting hole of the stationary member.

11 Claims, 7 Drawing Sheets



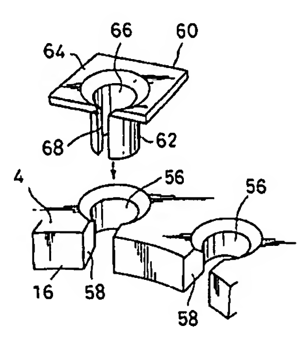


Fig.1

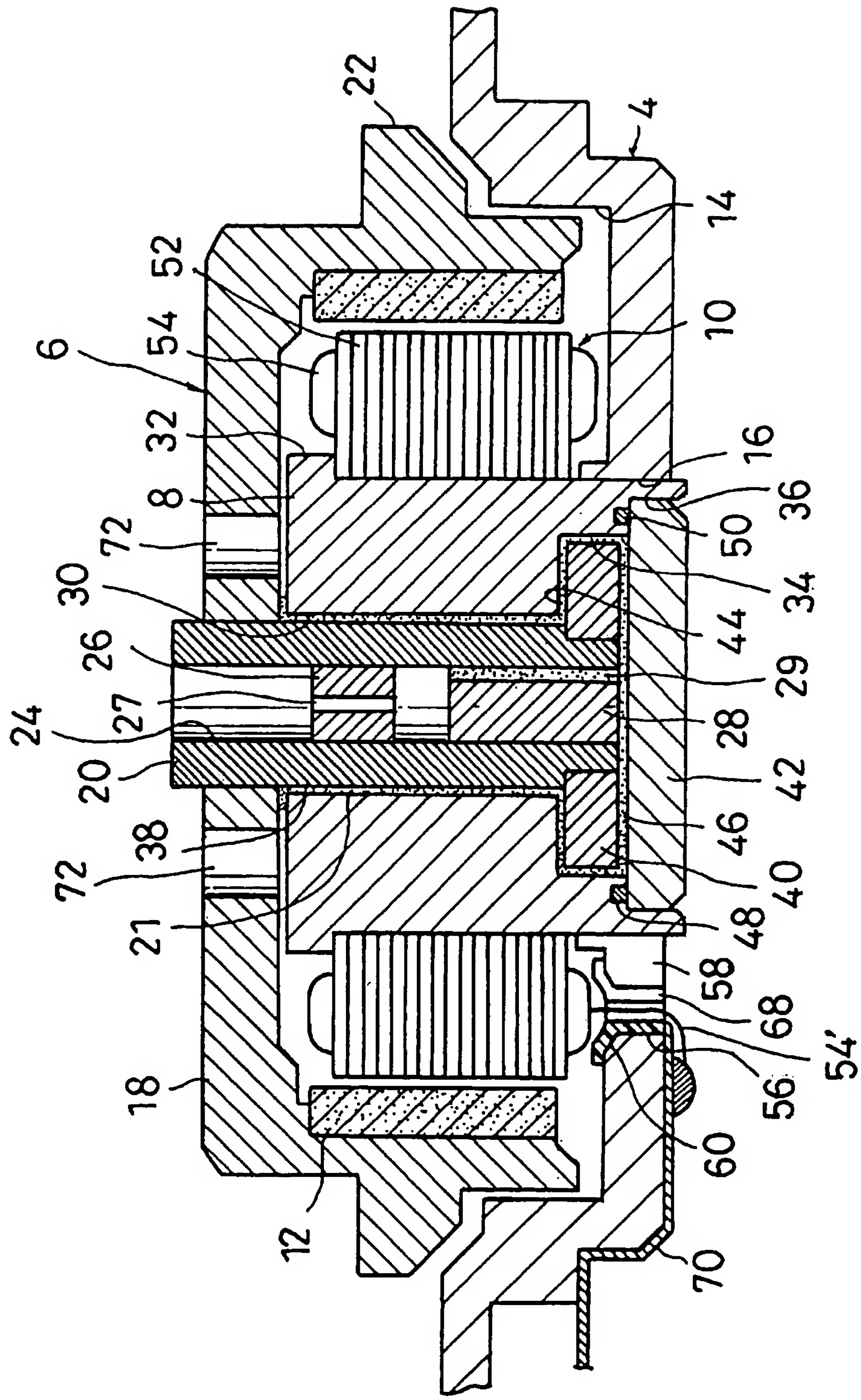

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Fig.2

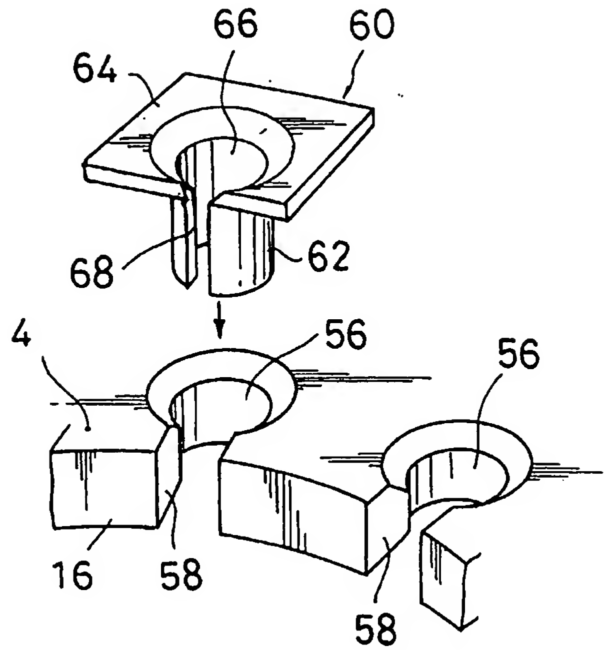

Fig.3

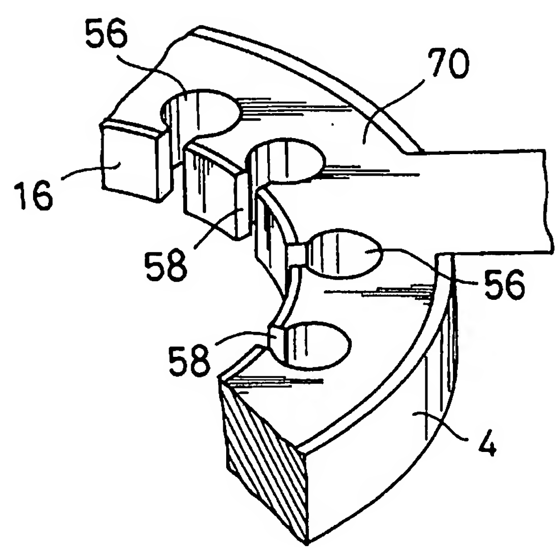

Fig.4

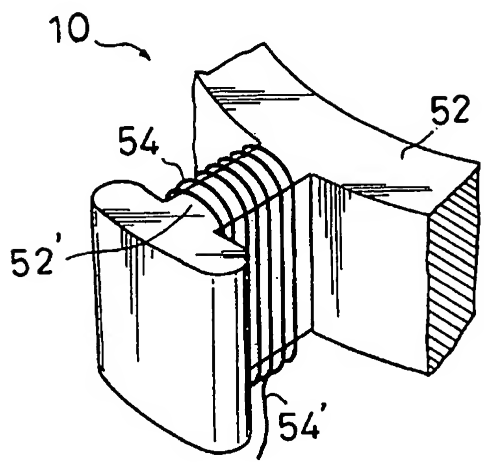

Fig.5

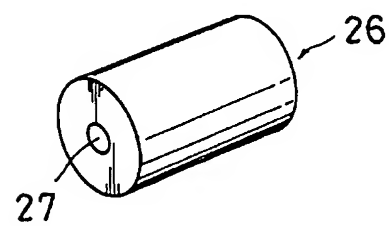

Fig.6

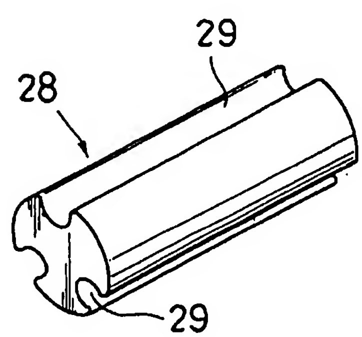

Fig.7

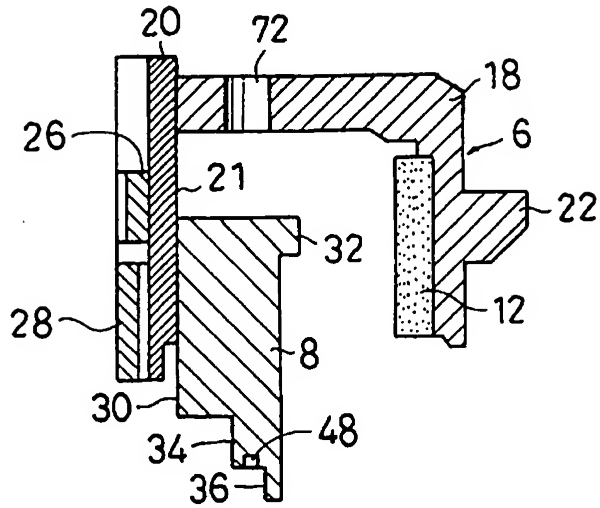

Fig.8

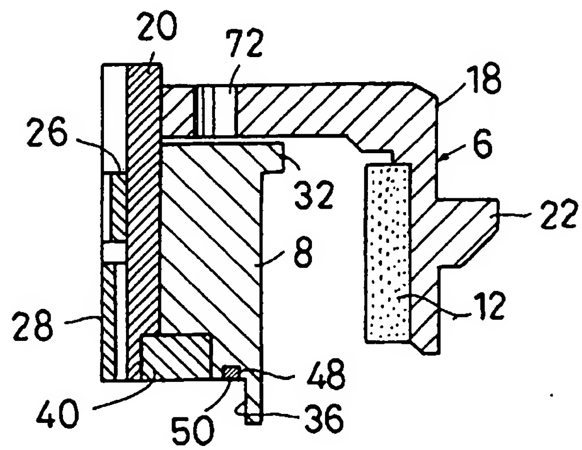

Fig.9

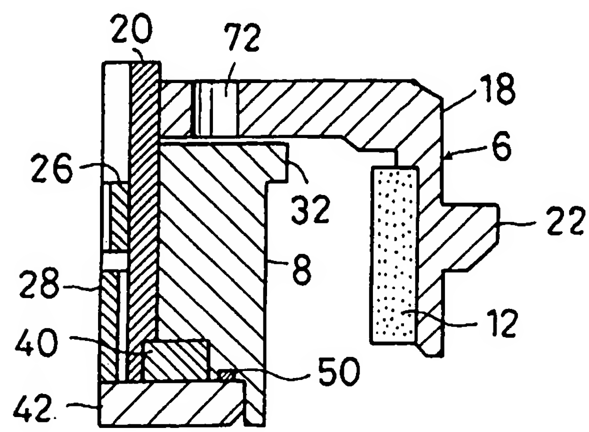

Fig.10

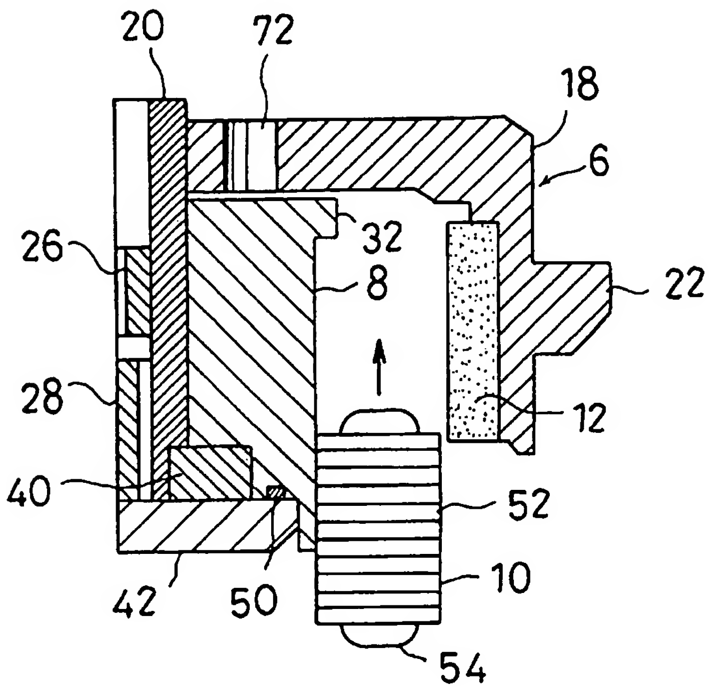
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Fig.11

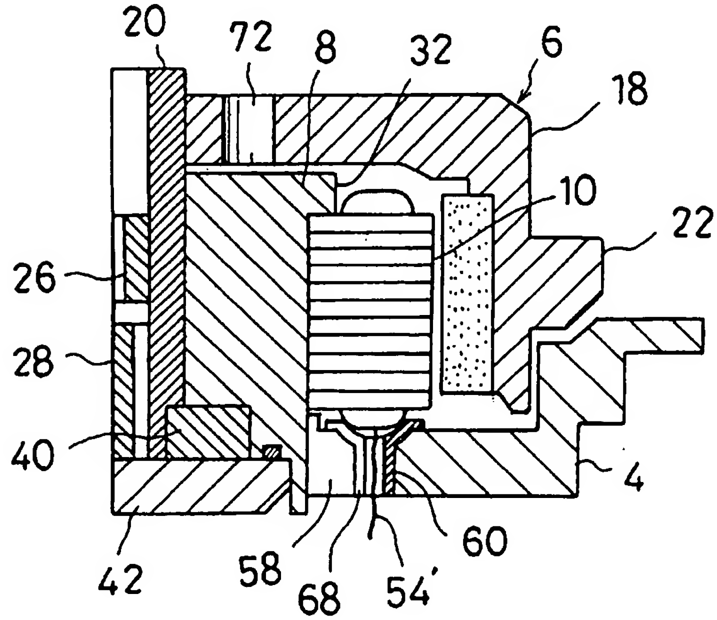
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Fig.12

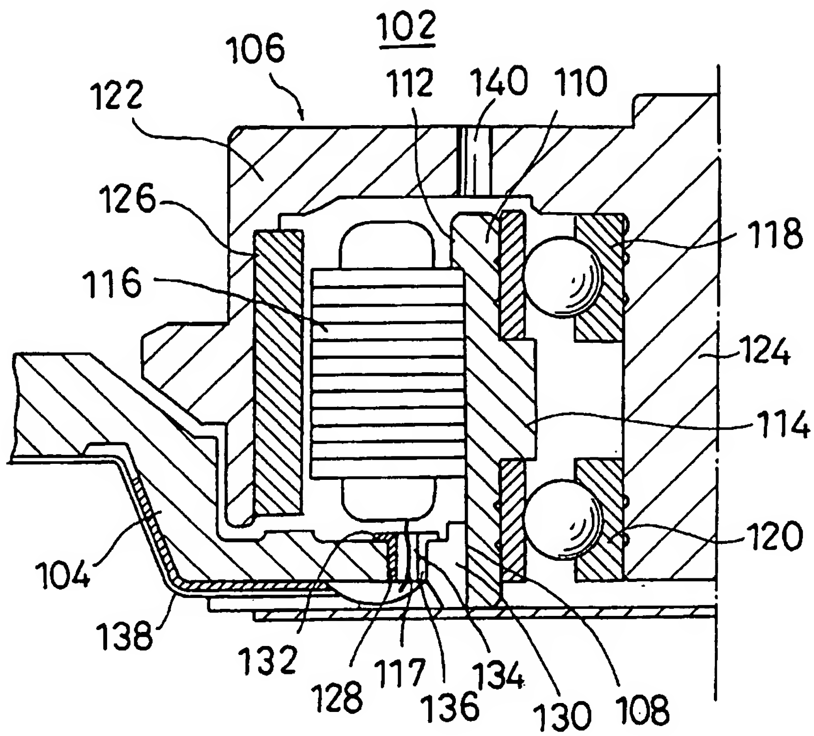
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Fig.13

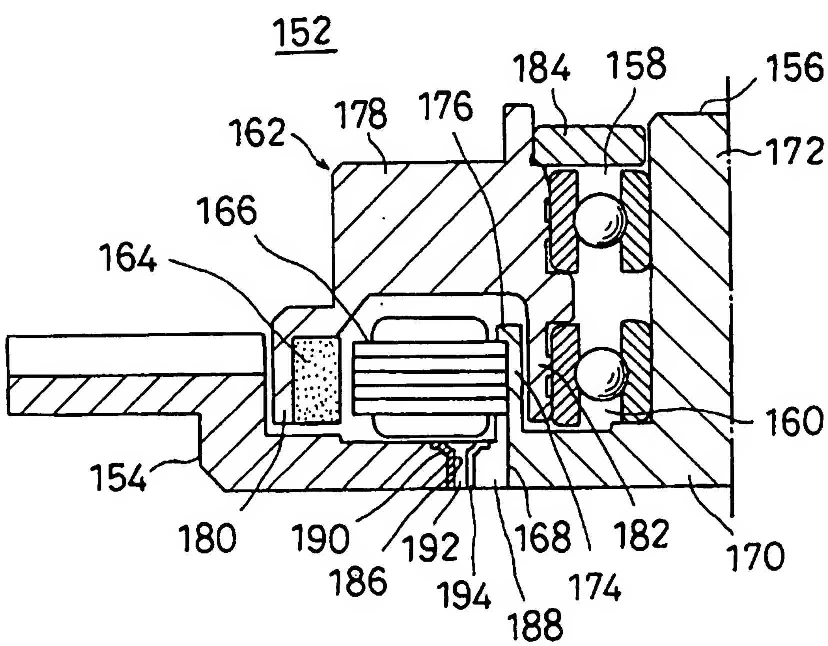
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Fig.14 (Prior Art)

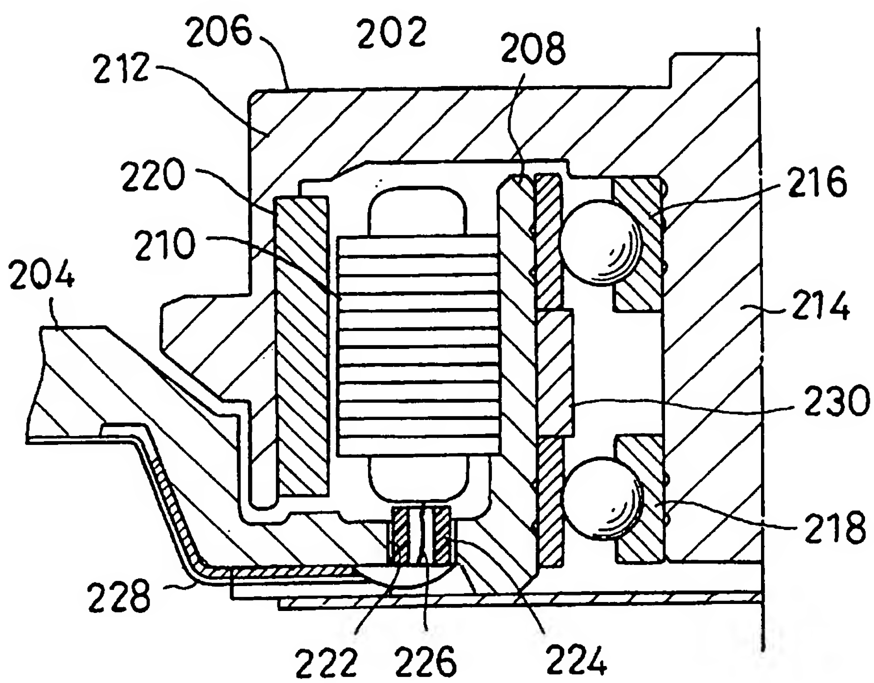
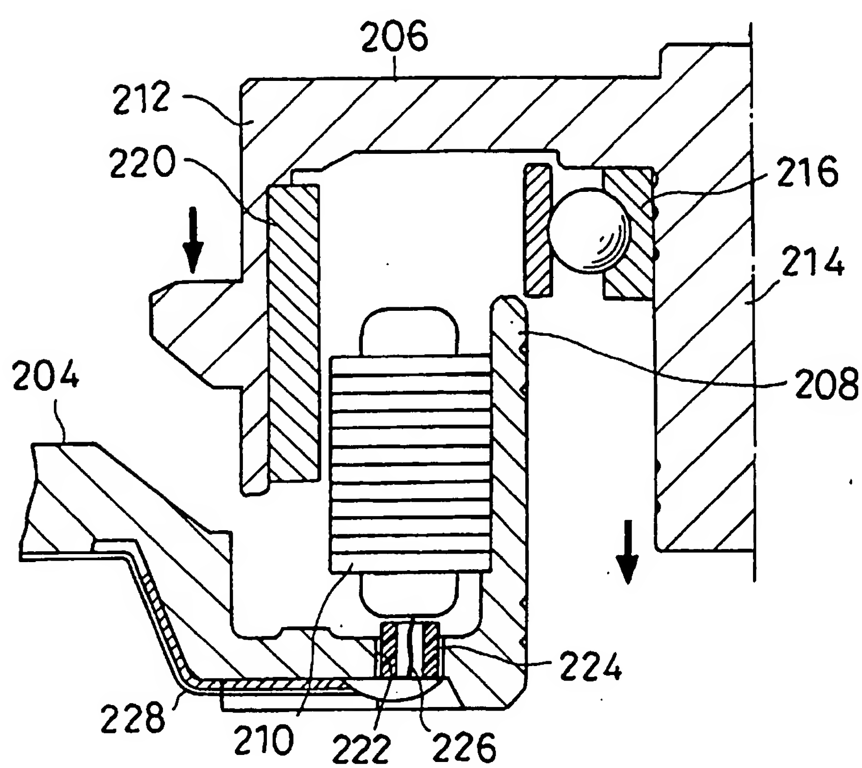


Fig.15 (Prior Art)



SPINDLE MOTOR DESIGN FACILITATING AND ENSURING PRECISE ASSEMBLY OF THE MOTOR

FIELD OF THE INVENTION

The present invention relates to a spindle motor for rotating disk recording mediums such as harddisks or optical disks.

DESCRIPTION OF THE PRIOR ART

A known spindle motor of a shaft rotation type for driving magnetic disks is constructed as illustrated in FIG. **14**. As shown, a spindle motor **202** comprises a substantially circular bracket **204** to be mounted to a disk drive device, and a rotor **206** rotatable relative to the bracket **204**. The bracket **204** is provided with a bearing support **208** at the cental portion thereof. A stator **210** is fitted onto the outer side of the bearing support **208**. The rotor **206** has a cup-like hub **212** on which a magnetic disk is mounted and a shaft **214** depending from the center of the hub **212**. The shaft **214** is rotatably mounted by a pair of ball bearings **216** and **218** to the inner side of the bearing support **208**. An annular rotor magnet **220** is fixedly mounted on the inner side of a circumferential wall of the hub **212** in such a way that it is located opposite to and spaced by a small gap from the outer periphery of the stator **210**. The bracket **204** is formed with through holes **222** at the positions opposite to the stator **210**. A tubular lead bushing **224** made of an insulating material is accommodated in each of the through holes **222**. Each lead wire **226** of the coils of the stator **210** is threaded through the lead bushing **224**, led beneath the bracket **204**, and welded by soldering to a given location on a flexible printed circuit board **228** which is bonded to the lower side of the bracket **204**. A spacer **230** is interposed between the outer races of the two ball bearings **216** and **218**.

The known spindle motor **202** is assembled by the following procedure. First, the lead bushings **224** are inserted into the through holes **222** of the bracket **204** and the flexible printed circuit board **228** is bonded to the lower side of the bracket **204**, as shown in FIG. **15**. The stator **210** is then lowered from above to fit onto the bearing support **208** of the bracket **204** and its coil leads **226** are threaded through the lead bushings **224** and soldered to the flexible printed circuit board **228**. Also, the rotor magnet **220** is fixedly mounted to the inner side of the outer wall of the hub **212** of the rotor **206**. The inner race of the upper bearing **216** is fitted onto a proximal end of the shaft **214** of the rotor **206**. This is followed by insertion of the shaft **214** of the rotor **206** together with the upper bearing **216** into the bearing support **208** of the bracket **204**, as shown in FIG. **15**, until the outer side of the outer race of the upper bearing **216** is directly joined to the uppermost of the inner side of the bearing support **208**. The spacer **230** and the lower bearing **218** are inserted from below to fit into the bearing support **208**. While the inner race of the lower bearing **218** is urged upwardly under pressure, the lower bearing **218** is secured with its inner race to the end portion of the outer side of the shaft **214** and with its outer race to the lowermost portion of the inner side of the bearing support **218**.

However, the above described structure of the spindle motor **202** has drawback in that it is susceptible to misalignment of parts in its assembly.

As seen from FIG. **15**, when the rotor **206** is being mounted to the bracket **204**, magnetic attractive force arises between the rotor magnet **220** and the stator **210** and may cause to tilt resulting in abutment of the rotor magnet **220** against the stator **210**. If the rotor **206** this occurs, the rotor may vibrates securely during rotation. Also, the space between the rotor magnet **220** and the stator **210** may vary causing the bearings to receive uneven stresses and disabling stable rotation.

In addition, it is troublesome to thread the coil leads **226** of the stator **210** through the hole of the lead bushings **224**. The smaller the size of the spindle motor, the more the job takes a time.

SUMMARY OF THE INVENTION

It is a primary object of the present invention to provide a motor structure which can be assembled smoothly and accurately.

It is another object of the present invention to provide a spindle motor which is free from mis-alignment due to tilting of a rotor relative to a stator during assembly.

It is a further object of the present invention to provide a spindle motor which can be assembled without being affected by the magnetic force between a rotor magnet and a stator coil.

It is a still further obejct of the present invention to provide a motor with which a lead wire derived from the coil of the stator can be easily threaded through a stationary bracket during the assembly of the motor.

Other objects and features of the present invention will be understood from the following description.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. **1** is a schematic cross sectional view of the entire construction of a spindle motor according to a first embodiment of the present invention;

FIG. **2** is a partially exploded perspective view of a lead bushing and a corresponding bushing hole provided on a bracket of the spindle motor shown in FIG. **1**;

FIG. **3** is a perspective view pf a part of the bracket of the spindle motor shown in FIG. **1**;

FIG. **4** is a partial perspective view of a part of a stator of the spindle motor shown in FIG. **1**;

FIG. **5** is a perspective view of a shaft sleeve of the spindle motor shown in FIG. **1**;

FIG. **6** is a perspective view of another type of shaft sleeve to be used in the spindle motor shown in FIG. **1**;

FIG. **7** is a partial cross sectional view showing a step for coupling a sleeve to a rotor in the spindle motor shown in FIG. **1**;

FIG. **8** is a partial cross sectional view showing a step for installing a thrust plate in the spindle motor shown in FIG. **1**;

FIG. **9** is a partial cross sectional view showing a step for installing a thrust cover in the spindle motor shown in FIG. **1**;

FIG. **10** is a partial cross sectional view showing a step for installing the stator in the spindle motor of FIG. **1**;

FIG. **11** is a partial cross sectional view showing a step for installing the bracket in the spindle motor of FIG. **1**;

FIG. **12** is a partial cross sectional view of a spindle motor according to a second embodiment of the present invention;

FIG. **13** is a partial cross sectional view of a spindle motor according to a third embodiment of the present invention;

FIG. **14** is a partial cross sectional view of a conventional spindle motor; and

FIG. **15** is a partial cross sectional view showing a step for assembling the spindle motor of FIG. **14**.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

The present invention will now be described in more detail referring to the accompanying drawings.

First, explanation will be made about a spindle motor according to a first embodiment of the present invention with reference to FIGS. 1 to 11.

The spindle motor of the first embodiment is of a shaft rotation type having dynamic pressure bearing means. As shown in FIG. 1, a spindle motor 2 comprises a bracket 4 which is to be mounted on a disk drive device (not shown). A rotor 6 is arranged for rotation relative to the bracket 4. A tubular sleeve 8 fixedly mounted on the bracket 4 serves as a part of the dynamic pressure bearing means for supporting the rotor 6. A stator 10 is fixedly mounted on the sleeve 8, and an annular rotor magnet 12 is integrally coupled to the rotor 6.

The bracket 4 is substantially of a disk shape having a circular recess 14 which is concentric with the motor axis. A mounting hole 16 is provided in the center of the bracket 4 coaxially with the recess 14. The sleeve 8 is securely fitted at its lower end to the mounting hole 16 so that it stands upright on the bracket 4.

The rotor 6 comprises a rotor hub 18 made of a magnetic material such as stainless steel and a tubular shaft 20 fixed coaxially to the rotor hub 18 and made of e.g. stainless steel. The rotor hub 18 has an inverted cup-like shape opening downwardly with the shaft 20 projecting in the opening. The rotor magnet 12 is bonded by adhesive to the inner side of a circumferential wall of the rotor hub 18. The outer side of the circumferential wall of the rotor hub 18 is shaped to hold a magnetic disk (not shown). More specifically, the magnetic disk is held on a flange 22 provided on the circumferential wall of the rotor hub 18. The outer side of the tubular shaft 20 is finished very precisely to have a smooth bearing surface 21. The shaft 20 has a central bore 24 therein into which a first shaft sleeve 26 (best shown in FIG. 5) and a second shaft sleeve 28 (best shown in FIG. 6) are tightly fitted. The first sleeve 26 is located at the middle portion of the shaft 20 and has a through hole 27 axially formed in the center thereof for communicating to the outside air. The second shaft sleeve 28 is located at the lower portion of the shaft 20 and has a plurality (three in this embodiment) of axially extending, angularly equally spaced slots 29 on the outer surface thereof.

The sleeve 8 comprises a tubular member made of a copper alloy and having an inner surface which is finished highly precisely to provide a smooth bearing surface 30. The outer side of the sleeve 8 has a flange-like projection 32 provided on the uppermost thereof. The sleeve 8 is formed with a thrust recess 34 in a lower region of the inner side thereof. The thrust recess 34 has a greater diameter than the inner bearing surface 30. The sleeve 8 is formed with a cover receiving recess 36 provided below the thrust recess 34 at the lower end of the inner side of the sleeve 8. The cover receiving recess 36 has a greater inner diameter than the thrust recess 34. A predetermined small gap 38 is formed or left between the inner bearing surface 30 of the sleeve 8 and the outer bearing surface 21 of the shaft 20. The gap 38 is filled with an appropriate fluid lubricant. A herringbone groove (not shown) for producing a dynamic pressure is provided on either the inner bearing surface 30 of the sleeve 8 and/or the outer bearing surface 21 of the shaft 20. This permits the shaft 20 to be radially within the sleeve 8 by means of the dynamic pressure in the sleeve 8.

An annular thrust plate 40 is fixed to the lower end of the shaft 20 and accommodated in the thrust recess 34 of the sleeve 8. A disk-like thrust cover 42 is fitted to the cover accepting recess 36 of the sleeve 8 thus closing the inner hollow of the tubular sleeve 8 at its lower end. A small gap 44 is formed between the upper side of the thrust plate 40 and a lower surface of the sleeve 8 exposed to the thrust recess 34. Also, a small gap 46 is formed between the lower side of the thrust plate 40 and the upper side of the thrust cover 42. Both the gaps 44 and 46 are filled with the fluid lubricant. A herringbone or spiral groove for producing a dynamic pressure is formed on both or either of the two opposite surfaces that define the small gaps 44 and 46. This allows the shaft 20 to be supported by the fluid lubricant with pressure being caused therein by the rotation of the thrust plate relative to the sleeve 8 with the thrust plate being restrained at a given axial position. In addition, an annular groove 48 is formed on a lower surface of the sleeve 8 exposed to the cover accepting recess 36. An 0-ring 50 is fitted into the annular recess 48 for airtight sealing of the interface between the sleeve 8 and the thrust cover 42.

As the second shaft sleeve 28 is located at the lower end position of the center bore 24 of the shaft 20, its slots 29 axially communicate the center bore 24 of the shaft 20 with the gap 44. The slots 29 of the second sleeve 28 are capable of holding an excess of the fluid lubricant lifted up by capillary action from the small gap 44 and thus act as a fluid lubricant storage. If required, the excess of the fluid lubricant stored in the slots 29 is fed to the dynamic pressure bearing means. As the through hole 27 of the first shaft sleeve 26 communicates the interior of the shaft 20 to the outside or atmosphere, the fluid lubricant is fed smoothly from the slots 29 to the dynamic pressure bearing means. Instead of forming the through hole 27 in the first shaft sleeve 26, a communicating aperture may be formed in a screw or retaining member (not shown) which is provided for fastening the upper end of the shaft 20. The aperture may not necessarily be a form of central hole but may be a slit or notch formed on the periphery of the screw or other retaining member. The slots 29 of the second shaft sleeve 28 are not limited to the configurations of the embodiment but may take any form so far as it functions as axially communicating channels in the outer side. The slots 29 of the second shaft sleeve 28 of FIG. 6 are preferably arranged in symmetry about the axis of rotation for balancing in the rotating movement of the shaft.

The stator 10 comprises, as shown in FIG. 4, a core 52 fixedly fitted on the outer side of the sleeve 8 and coils 54 wound on teeth 52' of the core 52. A coil lead 54' is derived from each of the coils 54. The stator 10 is radially spaced by a small gap from and arranged opposite to the rotor magnet 12.

The bracket 4 is formed with a group of coil lead passing slots 56 arranged at equal angular intervals adjacent to the mounting hole 16. Each of the coil lead passing slots 56 is communicated by a slit 58 to the mounting hole 16. A lead bush 60 which is made of an insulating material is fitted from above into the coil lead slot 56. The lead bush 60 comprises a tubular portion 62 fitted into the coil lead slot 56 and a flange-like retaining portion 64 formed integrally with the tubular member 62 at the top thereof. The tubular portion 62 allows the coil lead then lowered from above to fit onto the bearing support 208 54' to pass through a center hollow 66. The center hollow 66 is communicated with an axially extending opening 68 formed on the periphery of the tubular portion 62. The lead bush 60 is installed such that its opening 68 aligns with (the corresponding slit 58 of the bracket 4. Also, a flexible printed circuit board 70 is bonded to the lower side of the bracket 4. The coil leads 54' are threaded

through their corresponding lead bushes 60 and are connected by soldering to corresponding terminals on the flexible printed circuit board 70.

A procedure of assembling the spindle motor 2 of the first embodiment will now be explained referring to FIGS. 7 to 11. The procedure starts with inserting the first shaft sleeve 26 and the second shaft sleeve 28 into the center hollow 24 of the tubular shaft 20 and joining the shaft 20 to the rotary hub 18 as shown in FIG. 7. The rotor magnet 12 is then fixedly mounted to the inner side of the circumferential wall of the rotor hub 18 in order to complete the rotor 6. The shaft 20 of the rotor 6 is fitted into the sleeve 8.

Then, the thrust plate 40 is fixed to the lower end of the shaft 20 to restrain the sleeve 8, as shown in FIG. 8. At that time, the thrust plate 40 is seated in the thrust recess 34 of the sleeve 8. The O-ring 50 is also fitted into the annular groove 48 of the sleeve 8. This is followed by inserting the thrust cover 42 to a predetermined location in the cover accepting recess 36 of the sleeve 8 and securing it as shown in FIG. 9. Meanwhile, the small gaps 38, 44, and 46 are filled with the fluid lubricant prior to the installation of the thrust cover 42. This allows the rotor hub 18 to be rotatably supported by the dynamic pressure bearing means. The stator 10 with the coils 54 wound on its core 52 is then fitted onto the sleeve 8, as shown FIG. 10. When the upper side of the stator 10 has come into direct contact with the lower side of the flange-like projection 32 of the sleeve 8, the stator 10 is axially located in a correct position. Accordingly, the rotor magnet 12 on the rotor hub 18 is prevented from adverse engagement with the stator 10 during the installation regardless of magnetic interference between the rotor magnet 12 and the stator 10 and will provide no deflection of the rotor hub 18. As the shaft 20 of the rotor 6 is correctly fitted into the sleeve 8, the distance between the rotor magnet 12 and the stator 10 remains unchanged and uniform.

Meanwhile, lead bushes 60 are fitted into their corresponding coil lead passing slots 56 of the bracket 4 and the flexible printed circuit board 70 is bonded to the lower side of the bracket 4. Then, the lower end of the sleeve 8 carrying the rotor 6 is fitted into the mounting hole 16 of the bracket 4, as shown in FIG. 11. The fitting is carried out by pressing the sleeve 8 but not the rotor 6 towards the mounting hole 16. More particularly, a bar-like tool is inserted through a work aperture 72 provided in the rotor hub 18 opposite to the sleeve 8, and pressed against the sleeve 8 so that the sleeve 8 is fitted into the mounting hole 16. During the fitting of the sleeve 8 into the mounting hole 16, the coil leads 54' extending from the stator 10 across the mounting hole 16 which is communicated with the lead passing slots 56 of the bracket 4 and the lead bushes 60 installed in the slots 56. The coil leads are shifted from the mounting hole 16 through the slots 58 and the axial openings 68 into the center hollows 66 of the corresponding lead bushes 60. As the coil leads 54' pass through the center hollows 66 of the corresponding lead bushings 60, they are easily drawn out from the bracket 4. The coil leads 54' are finally connected by soldering to the flexible printed circuit board 70.

The spindle motor 2 of the first embodiment has the following significant advantages.

When the stator 10 is mounted to the sleeve 8 which services as a stationary member, the stator 10 is inserted axially and inwardly from the lower opening of the rotor hub 18 with the stator being moved along the outer side of the sleeve 8 and secured at a position upon its coming into contact with the flange-like projection 32. This allows the stator 10 to be affected as little as possible by the magnetic interference between the rotor hub 18 and the rotor magnet 12 on the rotor hub 18 while the stator 10 is being installed and anchored to the stationary member or sleeve 8, thereby insuring precise assembly. The stator 10 is to be mounted on the sleeve 8 which has been coupled with the rotor 6 in advance to support it. The rotor 6 is hence free from the magnetic interference and remains for highly accurate rotation since the rotor 6 has been supported precisely even if the stator 10 is affected by the magnetic interference of the rotor hub 18 and the rotor magnet 12 on the rotor hub 18. The coil leads 54' of the stator 10 are easily drawn out because the lead passing slots 56 of the bracket 4 and the lead bushings 60 installed in the slots 56 are communicated through the slots 58 and the axial openings 68 to the mounting hole 16. This will increase the quality and efficiency of assembly work. Particularly, this embodiment employs the dynamic bearing means provided with the storage of the fluid lubricant and will provide highly accurate, life-long bearing functions and thus smooth rotating actions.

A spindle motor according to a second embodiment of the present invention will be described referring to FIG. 12.

The second embodiment is the spindle motor 102 of shaft rotation type using ball bearings as the bearing means. The spindle motor 102 includes a substantially circular bracket 104 mounted to a disk drive and a rotor 106 arranged for rotation in relation to the bracket 104. The bracket 104 has a round mounting hole 108 provided in the center thereof and a separate tubular bearing support 110 is fitted at the lower end into the mounting hole 108. The bearing support 110 has a flange-like projection 112 provided on the upper end of the outer side thereof and an inwardly projecting land 114 provided on a center of the inner side thereof. A stator 116 is fitted onto the outer side of the bearing support 110 as positioned with the flange-like projection 112. A pair of ball bearings 118 and 120 are fitted to the inner side of the bearing support 110 at both upper and lower ends respectively as positioned with the land 114.

The rotor 106 comprises a cup-like rotor hub 122 on which a magnetic disk is mounted and a shaft 124 extending from the center of the rotor hub 122. The shaft 124 is rotatably joined by a pair of ball bearings 118 and 120 to the inner side of the bearing support 110. An annular rotor magnet 126 is fixedly mounted to the inner side of a circumferential wall of the rotor hub 122. The rotor magnet 126 is spaced by a small gap from and located opposite to the outer side of the stator 116.

A group of lead passing slots 128 corresponding to coil leads 117 from the stator 116 are provided in the bracket 104 at equal intervals about and adjacent to the mounting hole 108. Each of the lead passing slots 128 is communicated through an axially extending slit 130 to the mounting hole 108. A lead bushing 132 made of an insulating material is fitted into the lead passing slot 128. The lead bushing 132 comprises a tubular body inserted into the lead passing slot 128 and a flange provided at the upper end of the tubular body. A center hollow 134 is provide across the tubular body. The lead bushing 132 has an axially extending opening 136 provided in the tubular body thereof. The center hollow 134 is hence communicated through the opening 136 and the slit 130 to the mounting hole 108 of the bracket 104. A flexible printed circuit board 138 is bonded to the lower side of the bracket 104.

A procedure of assembling the spindle motor 102 of the second embodiment will now be explained.

The procedure starts with mounting the rotor magnet 126 to the rotor hub 122, fitting the inner race of the upper ball

bearing 118 onto the upper end of the shaft 124, and inserting the shaft 124 from above into the bearing support 110 to join the outer race of the upper bearing 118 directly to the bearing support 110. The lower bearing 120 is inserted from below into the bearing support 110 so that its outer race is directly joined to the bearing support 110 and its inner race to the lower end of the shaft 124. The two ball bearings 118 and 120 are bonded by means of adhesives under pressure. Accordingly, the rotor 106 is rotatably linked by the two ball bearings 118 and 120 to the bearing support 110.

This is followed by fitting from the opening of the rotor hub 122 the stator 116 onto the outer side of the bearing support 110 carrying the rotor 106. As the stator 116 is inserted along the outer side of the bearing support 110 into the rotor 106 which remains rotatably supported with the bearing support 110 without any deflection, it is prevented from directly engaging with the rotor magnet 126. This will minimize the magnetic interference between the stator 116 and the rotor magnet 126. The stator 116 is positioned when it comes into contact with the flange-like projection 112 of the bearing support 110.

Meanwhile, the lead bushings 132 are fitted into their respective lead passing slots 128 of the bracket 104 and the flexible printed circuit board 138 is bonded to the lower side of the bracket 104. Then, the lower end of the bearing support 110 carrying the rotor 106 is fitted under pressure into the mounting hole 108 of the bracket 104. The pressure fitting of the bearing support 110 into the mounting hole 108 is carried out by inserting a bar-like tool into a work hole 140 provided in the upper side of the rotor hub 122 and pressing it against the bearing support 110. Before the fitting of the bearing support 110 into the mounting hole 108, the coil leads 117 extending from the stator 116 across the mounting hole 108 are shifted from the mounting hole 108 through the slots 130 and the axial openings 136 into the center hollows 134 of the corresponding lead bushings 132 for drawing out. When the bearing support 110 has been fitted into the mounting hole 108, the coil leads 117 are connected by soldering to the flexible printed circuit board 138.

In the spindle motor 102 of the second embodiment similar to the first embodiment, the stator 116 and the rotor 106 are coupled to each other without direct engagement between the stator 116 and the rotor magnet 126 hence increasing the quality and efficiency of assembly operation. Also, the coil leads 117 of the stator 116 can easily be drawn out thus encouraging the assembly operation.

A spindle motor according to a third embodiment of the present invention will be described referring to FIG. 13.

The spindle motor 152 of the third embodiment is of a stationary shaft type using ball bearing means. The spindle motor 152 includes a substantially circular bracket 154 mounted to a disk drive, a stationary shaft 156 provided integral with the bracket 154 in the center, a rotor 162 rotatably supported via a pair of ball bearings 158 and 160 by the shaft 156, an annular rotor magnet 164 mounted to the rotor 162, and a stator 166 provided integral with the bracket 154. The outer side of the stator 166 is spaced by a small gap from and located opposite to the inner side of the rotor magnet 164.

The bracket 104 has a round mounting hole 168 provided in the center thereof. The shaft 156 comprises a disk-shaped bottom plate 170, a capstan 172 vertically extending from the center of the bottom plate 170, and a cylinder 174 extending upwardly from the outer edge of the bottom plate 170. The bottom plate 170 is fitted into the mounting hole 168 of the bracket 154. A flange-like projection 176 is provided on the upper end of the cylinder 174.

The rotor 162 includes a rotor hub 178 made of a magnetic material such as stainless steel and an outer cylinder 180 extending downwardly from the outer edge of the rotor hub 178. A rotor magnet 164 is bonded by adhesive to the inner side of the outer cylinder 180. The upper ball bearing 158 is mounted to the inner side of the rotor hub 178 and the lower ball bearing 160 to the inner side of an inner cylinder 182 extending downwardly from the inner edge of the rotor hub 178. An annular labyrinth cap 184 is fitted into the upper end of the inner side of the rotor hub 178. The inner side of the labyrinth cap 184 is spaced by a small gap from and located opposite to the upper end of the outer side of the stationary shaft 172 hence forming a labyrinth sealing structure. The labyrinth cap 184 is located above the upper ball bearing 158 or at the motor outward side for preventing unwanted dirt of e.g. grease from escaping from the ball bearings 158 and 160 to the outside of the motor 152. The inner cylinder 182 of the rotor hub 178 is radially spaced by a small gap from and arranged opposite to the cylinder 174 of the shaft 156, forming another labyrinth sealing structure.

A group of lead passing slots 186 are provided in the bracket 154 about and adjacent to the mounting hole 168. Each of the lead passing slots 186 is communicated through a slit 188 to the mounting hole 168. A lead bushing 190 made of an insulating material is fitted into the lead passing slot 186. The lead bushing 190 has an axially extending center hollow 192 provided therein and an axially extending opening 194 thereof so that the center hollow 192 is communicated with the slit 188.

A procedure of assembling the spindle motor 152 of the third embodiment will now be explained.

The procedure starts with mounting the rotor magnet 164 to the rotor hub 178 and rotatably joining the shaft 156 by the two ball bearings 158 and 160 to the stationary shaft 172.

Then, the stator 166 is inserted from the opening side of the rotor hub 178 and fitted onto the outer side of the cylinder 174 of the shaft 156 carrying the rotor 162. As the stator 166 is inserted along the outer side of the shaft 156 into the rotor 162 which remains supported securely with the shaft 156 with no deflection, it is prevented from direct engagement with the rotor magnet 164. This will minimize the magnetic interference between the stator 166 and the rotor magnet 166. The stator 166 is positioned when it comes into direct contact with the flange-like projection 176 of the cylinder 174.

Meanwhile, the lead bushings 190 are fitted into their respective lead passing slots 186 of the bracket 154. The lower end or the outer side of the bottom plate 170 of the shaft 156 carrying the rotor 162 is fitted under pressure into the mounting hole 168 of the bracket 154. The pressure fitting of the shaft 156 into the mounting hole 168 is conducted by directly pressing the upper side of the stationary shaft 172. Before the fitting the shaft 156 into the mounting hole 168, coil leads extending from the stator 166 are shifted from the mounting hole 168 through the slits 188 and the openings 194 into the center hollows 192 of the lead bushings 190 for drawing out.

In the spindle motor 152 of the third embodiment similar to the first or second embodiment, the stator 166 and the rotor 162 are coupled to each other without direct engagement between the stator 166 and the rotor magnet 164 hence increasing the quality and efficiency of assembly operation. Also, the coil leads of the stator 166 can easily be drawn out thus encouraging the assembly operation.

It is understood that the present invention is not limited to the spindle motors of the foregoing embodiments but other

changes and modifications are possible without departing the scope of the present invention. For example, the present invention is applicable with equal success to a base-mounted type spindle motor in which a rotor is supportedly joined by bearing means to the base plate of a magnetic disk drive, as compared with the prescribed spindle motor of which bracket is mounted as the stationary member to the base plate of a magnetic disk drive.

What is claimed is:

1. A spindle motor having a stationary member, a bearing means mounted to the stationary member, a rotor rotatably coupled by the bearing means to the stationary member, an annular stator mounted to the stationary member radially outwardly of the bearing means, and an annular rotor magnet mounted to the rotor so that it is radially outwardly spaced by a small distance from and arranged opposite to the stator, wherein:

the stationary member includes a round mounting hole which is arranged coaxially with the rotor and the diameter which is equal to the inner diameter of the stator;

the stationary member also includes a tubular member which has an outer diameter substantially equal to the diameter of the mounting hole and which is coupled at its inner side with the bearing means so that the rotor is rotatably supported via the bearing means by the tubular member;

the stator is fitted onto the outer side of the tubular member which rotatably supports the rotor, and the tubular member is fitted at one end into the mounting hole and thus secured to the stationary member;

the bearing means comprises the tubular member and a dynamic pressure bearing defined by a shaft mounted in the interior thereof for rotation in relation thereto, either the inner side of the member or the outer side of the shaft having a groove for producing radial dynamic pressure;

the rotor comprises the shaft and an inverted a cup-like rotor hub fixedly mounted to the shaft with the rotor magnet joined to the inner side of a circumferential wall of the rotor;

the tubular member has a thrust recess with an inner diameter greater than the inner diameter of the interior of the member and a cover accepting recess with an inner diameter greater than the inner diameter of the thrust recess, both provided in the inner side thereof and aligned axially with each other, the cover accepting recess opening to the lower end of the tubular member; and

a thrust plate is fitted into the thrust recess and secured to the lower end of the shaft and a thrust cover is fitted into the cover accepting recess to close the tubular member; a groove for producing a dynamic thrust being provided either on both axial sides of the thrust plate or the inner sides of the tubular member and thrust cover.

2. A spindle motor comprising:

A rotatable shaft;

a cylindrical sleeve member surrounding the shaft and having a flange formed on an upper end portion of an outer peripheral surface thereof;

a rotor including a disk-shaped upper wall fixed to the shaft and having a plurality of access holes formed at portions axially opposing the sleeve member so as to expose a part of the top portion of the sleeve member, and a cylindrical wall depending from the outer peripheral portion of the upper wall;

a bearing means interposed between the sleeve member and the shaft so as to support the rotor for rotation;

an annular stator including a core having a plurality of teeth portions and a plurality of coils respectively wound around the teeth portions, the annular stator being mounted on the outer peripheral surface of the sleeve member so as to be in contact with the flange in an axial direction for positioning the stator;

an annular rotor magnet attached to the inner peripheral surface of the cylindrical wall of the rotor so that the stator and the rotor magnet face each other through a small radial gap;

a base member formed with a central opening for fixedly receiving the sleeve member coaxially with each other, the base member further provided with a plurality of slots formed in the vicinity of the central opening for receiving the wires of the coils and a plurality of slots communicating between the central opening and the slots;

a plurality of hollow tubular bush members made of an insulating resin and inserted respectively into the slots; and

a flexible printed circuit board attached to a lower surface of the base member, the wires of the coils being led from the stator through the tubular bush members in the slots and electrically connected to the printed circuit board.

3. A spindle motor according to claim 2, wherein each tubular bush member includes a hollow tubular portion and a flange integrally formed with the tubular portion at the top thereof, the bush members further include an opening aligning with the corresponding slit of the base member so as to communicate between an interior of the bush member and the central opening.

4. A spindle motor according to claim 2, wherein an outer peripheral surface of the rotational shaft and an inner peripheral surface of the sleeve member radially oppose each other with a narrow gap therebetween filled with lubricating fluid, a hydrodynamic pressure generating groove being formed on at least one opposing surface of the rotational shaft and the sleeve member for generating hydrodynamic pressure during rotation of the rotor for bearing radial load of the rotor;

a disk-like thrust plate fixed on a lower end portion of the rotatable shaft and a disk-like thrust bush fixedly fitted on the inner peripheral surface of the sleeve member so as to axially oppose the thrust plate through a narrow gap filled with lubricating fluid, a hydrodynamic pressure generating groove being formed on at least one opposing surface of the thrust plate and the thrust bush for generating hydrodynamic pressure during rotation of the rotor for bearing the thrust load of the rotor.

5. A spindle motor according to claim 4, wherein the rotatable shaft has a through hole extending axially and an oil reservoir is formed at one end portion of the hole so as to retain lubricating fluid by capillary action.

6. A spindle motor according to claim 5, wherein a columnar member having a plurality of slots extending in the axial direction is inserted in the through hole, a plurality of channels formed between an inner surface of the through hole and the slots is provided within the through hole so as to form the oil reservoir.

7. A spindle motor according to claim 6, wherein the small gaps between the inner side of the tubular member and the outer side of the shaft, between the thrust recess side of the tubular member and the thrust plate and the thrust cover are filled with a fluid lubricant.

8. A spindle motor according to claim 7, wherein the shaft has a hollow center provided at one side with a storage region for saving a portion of the fluid lubricant by the effect of capillary action.

9. A spindle motor according to claim 8, wherein a shaft sleeve is fitted into the center hollow of the shaft and a plurality of axially extending grooves are provided in either the inner side of the hollow shaft or the outer side of the shaft sleeve for holding a portion of the fluid lubricant by the capillary action.

10. A spindle motor according to claim 2, wherein a pair of ball bearings is interposed between an outer peripheral surface of the shaft and an inner peripheral surface of the sleeve member for rotatably supporting the rotational shaft.

11. A spindle motor according to claim 10, wherein an annular projection is formed on the inner peripheral surface of the sleeve member for positioning the ball bearings in the axial direction.

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UNITED STATES PATENT AND TRADEMARK OFFICE

CERTIFICATE OF CORRECTION

PATENT NO. : 6,097,121
DATED : August 1, 2000
INVENTOR(S) : Yoshito Oku

It is certified that error appears in the above-identified patent and that said Letters Patent is hereby corrected as shown below:

On the Title Page

[30] Foreign Application Priority Data
October 6, 1995 [JP] Japan 7-286484

[62] Related U.S. Application Data
Division of application No. 08/725,977, Oct. 4, 1996, Pat. No. 5,831,355

Signed and Sealed this

Twenty-fourth Day of April, 2001

Attest:

Nicholas P. Godici

NICHOLAS P. GODICI

Attesting Officer

Acting Director of the United States Patent and Trademark Office

United States Patent [19]
Tanaka et al.

[11] **Patent Number:** **4,805,972**
[45] **Date of Patent:** **Feb. 21, 1989**

[54] **DYNAMIC PRESSURE GAS BEARING DEVICE**

[75] Inventors: **Katsuhiko Tanaka**, Yamato-shi; **Takanobu Sato**, Odawara-shi; **Ikunori Sakatani**, Fujisawa-shi, all of Japan

[73] Assignee: **Nippon Seiko Kabushiki Kaisha**, Tokyo, Japan

[21] Appl. No.: **8,149**

[22] Filed: **Jan. 22, 1987**

Related U.S. Application Data

[63] Continuation of Ser. No. 810,530, Dec. 17, 1985, abandoned, which is a continuation of Ser. No. 463,011, Feb. 1, 1983, abandoned.

[30] **Foreign Application Priority Data**

Feb. 5, 1982 [JP] Japan 57-16299
May 14, 1982 [JP] Japan 57-80213

[51] **Int. Cl.⁴** **G02B 26/10; F16C 17/10**
[52] **U.S. Cl.** **350/6.7**; 310/90; 384/99; 384/107; 384/113; 384/124; 384/372
[58] **Field of Search** 305/6.5–6.8; 310/90, 157; 384/99, 100, 107, 108, 112–115, 121, 123, 368, 372, 373, 416, 124

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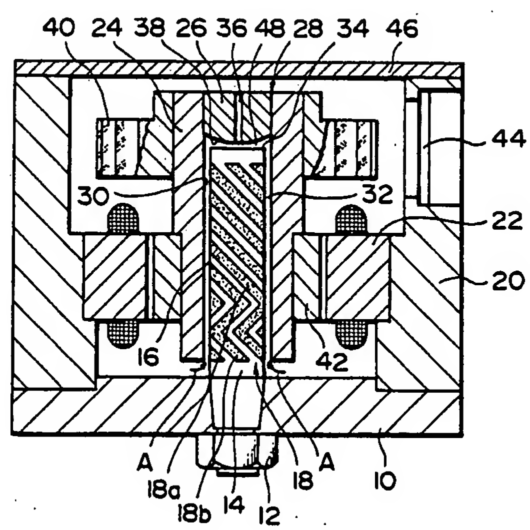
Primary Examiner—David Werner
Attorney, Agent, or Firm—Wyatt, Gerber Burke and Badie

[57] **ABSTRACT**

Disclosed is a dynamic pressure gas bearing device in a rotational unit in which a rotational member put on a cantilevered fixed shaft is designed such that an operating gas generated by a dynamic pressure groove formed between the fixed shaft and the rotational member is directed into a pressure chamber between the fixed shaft and the rotational member and supports the rotational member in the thrust direction and that the pressure in the pressure chamber is adjusted by a hole formed in the fixed shaft or the rotational member.

20 Claims, 2 Drawing Sheets



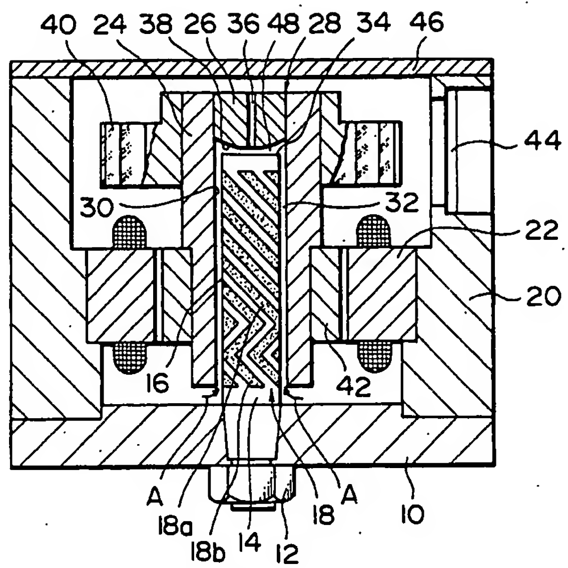

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FIG. 1

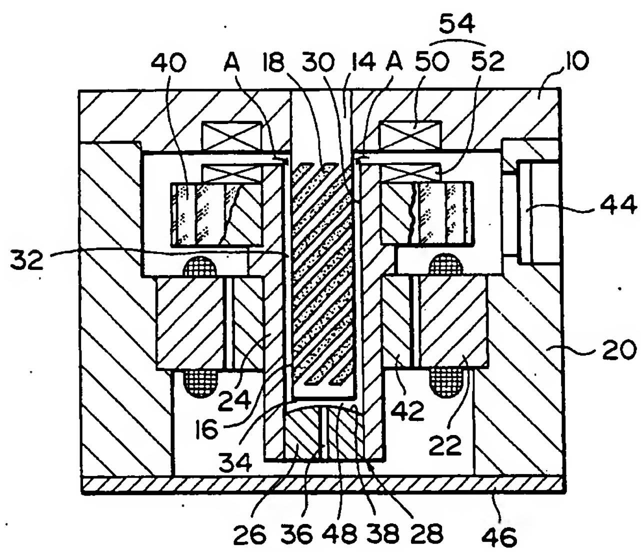
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FIG. 2

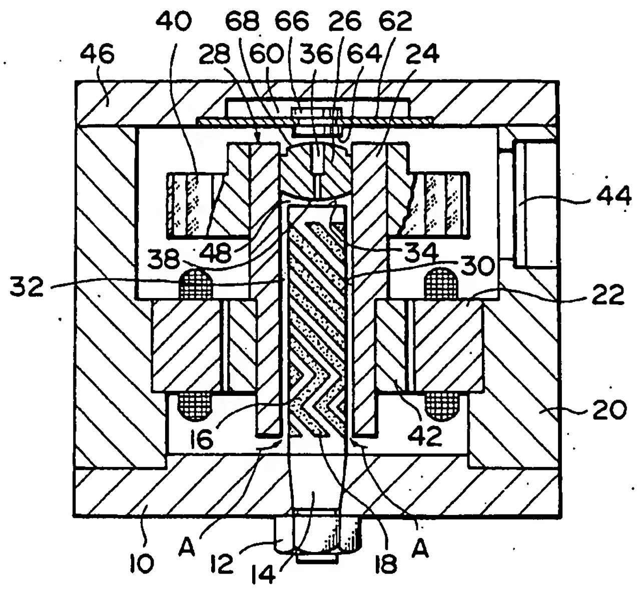
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FIG. 3

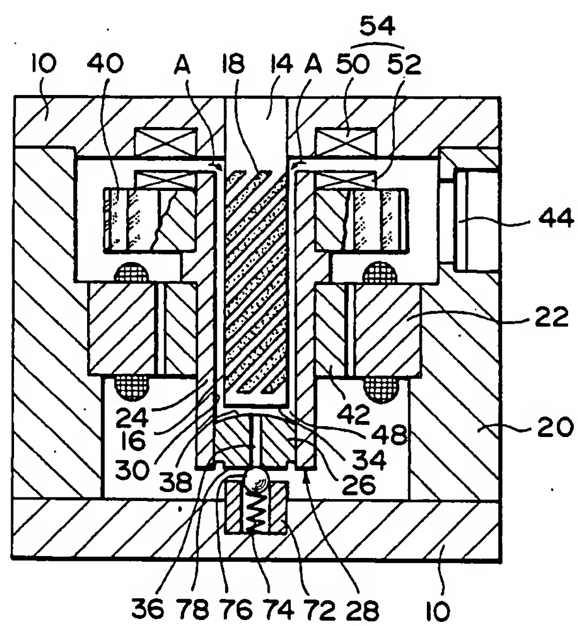
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FIG. 4

DYNAMIC PRESSURE GAS BEARING DEVICE

This application is a continuation application based upon Application Ser. No. 810,530, filed Dec. 17, 1985 and entitled "Dynamic Pressure Gas Bearing Device" now abandoned, which is a continuation of Application Ser. No. 463,011 filed Feb. 1, 1983, now abandoned.

BACKGROUND OF THE INVENTION

1. Field of the Invention

This invention relates to a dynamic pressure gas bearing device for use in the rotational unit of a business machine, an image instrument, an information instrument, an optical instrument or the like.

2. Description of the Prior Art

Heretofore, a structure in which the opposite ends of a shaft on which a polygon mirror is mounted are supported by ball bearings has often been used in a bearing device for the rotational unit of a rotational polygon mirror light deflector used in an instrument such as a laser beam printer. However, in a spindle device using the conventional ball bearings, it is not easy to attain the required accuracy of rotation, an irregularity of rotation results because of vibration based on the error in the machining of the race for the ball bearings, the vibration caused by the passage of the balls, and the vibration caused by the retainer or the irregularity of rotation caused by the grease enclosed in the ball bearings being irregularly bitten by the balls in rotation are unavoidable.

In such a rotational polygon mirror light deflector, where the accuracy of rotation of the polygon mirror in highspeed rotation is poor and where there is irregularity of rotation of the polygon mirror, the characters printed become blurred and, therefore, very severe accuracy of dynamic rotation is required of the supporting bearings. However, with the recent tendency of printers toward a higher speed and a smaller size, the number of revolutions of the rotational polygon mirror light deflector has increased from several thousand rpm to several tens of thousand rpm and the apparatus itself is in the tendency toward a smaller size. It has therefore become more and more difficult to improve the accuracy of rotation and eliminate the irregularity of rotation. Also, with the tendency toward a higher speed, the life of the ball bearings has become shorter and problems in reliability have arisen.

Further, to prevent the polygon mirror from being stained, it is desired that a lubricant such as grease which may scatter or evaporate not be used for the support bearings. However, the deterioration of the performance of the polygon mirror by scattering or evaporation of grease is unavoidable with the ball bearings because the ball bearings are greaselubricated. Even if magnetic fluid seal is used, scattering or evaporation of the oil itself used in the magnetic fluid is unavoidable and the use of such seal cannot be an essential countermeasure. Also, in the case of the ball bearings, prepressure adjustment has been necessary and therefore assembly has not always been easy, and it has been difficult from the viewpoint of mass production to assemble the ball bearings so that there is no mounting error to maintain dynamic rotational accuracy.

SUMMARY OF THE INVENTION

It is a first object of the present invention to provide a dynamic pressure gas bearing device which is improved in the dynamic rotational accuracy, the reliability during high-speed rotation and the durability.

It is a second object of the present invention to provide a dynamic pressure gas bearing device in which the starting torque of the rotational member is small and damaging of the various parts when the rotation is stopped is prevented.

It is a third object of the present invention to provide a dynamic pressure gas bearing device of which the environment is not contaminated but is kept clean during the use of the device.

It is a fourth object of the present invention to provide a dynamic pressure gas bearing device of which the entire structure is simplified and which is easy to assemble and excellent in mass productivity and which can be manufactured at low cost.

The invention will become fully apparent from the following detailed description thereof taken in conjunction with the accompanying drawings.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a front cross-sectional view showing a first embodiment of the present invention.

FIG. 2, 3 and 4 are front cross-sectional views showing second, third and fourth embodiments, respectively, of the present invention.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

Referring to FIG. 1, fixed shaft 14 is installed upright on a plate-like base bed 10 by means of a nut 12 and a dynamic pressure generating groove 18 is formed in the outer peripheral, surface 16 of the fixed shaft 14. The groove 18 comprises a spiral groove 18*a* and herring bone grooves 18*b*. A cylindrical housing 20 is also secured to the base bed 10 and the stator 22 of a drive motor is secured to the inner peripheral surface of the intermediate portion of the housing 20 in the direction of the height of the fixed shaft 4. A rotor 42 which will hereinafter be described is opposed to the stator 22.

A rotational member 28 comprising a sleeve 24 and a thrust bearing member 26 pressure-fitted into one of the openings of the sleeve 24 is loosely fitted to the fixed shaft 14. The inner peripheral surface of the sleeve 24 is opposed to the outer peripheral surface 16 of the fixed shaft 14 with a predetermined gap 32 left therebetween, and the upper surface thereof provides a flat thrust end surface 34.

An axially extending hole 36 is formed centrally through the thrust bearing member 26 and the lower surface of the thrust bearing member 26 is formed into a thrust end surface 38 comprising a convex spherical surface. At least one of the thrust end surfaces 34 and 38 should desirably be formed into a convex spherical surface to decrease the starting torque and reduce the damage when the rotation is stopped, and accordingly the thrust end surface 34 may also be formed into a convex spherical surface. However, it is more effective to form the axial hole 36 in the thrust end surface 38 forming a convex spherical surface as in the present embodiment because the convex spherical surface can be prevented from eating into the axial hole. Also, one thrust end surface may be formed into a convex spherical surface and the other cooperating thrust end surface may be formed into a concave spherical surface having a radius slightly greater than that of the convex spherical surface.

In this embodiment, a material such as plastics of good slidability or ceramics of good wear is used for the thrust bearing member 26 to decrease the starting torque on the thrust end surface 38 during the starting of the rotational member 28 and moreover improve the wear when the rotation is stopped. In the present embodiment, the sleeve 24 and the thrust bearing member 26 are discrete members, but alternatively, they may be formed integrally with each other by the same member.

A polygon mirror 40 is secured to the sleeve 24 toward the upper portion thereof and a rotor 42 opposed to and cooperating with the stator 22 is secured to the sleeve 24 toward the lower portion thereof.

Also, toward the upper portion of the housing 20, namely, at the level corresponding to the polygon mirror 40, there is provided a glass window 44 for passing a laser beam therethrough, and the upper opening of a housing 20 is covered by a cover 46.

The operation of the present embodiment having the above-described construction will now be described. During the repose of the rotational member 28 (including the time of low speed rotation thereof), the thrust end surface 38 of the thrust bearing member 26 is in contact with the thrust end surface 34 of the fixed shaft 14.

As the rotational member 28 is driven by the drive motor and rotated clockwisely as viewed from above in FIG. 1, the ambient gas advances in the direction of arrow A due to the action of the dynamic pressure generating groove 18 and flows into the gap 32 between the outer peripheral surface 16 of the fixed shaft forming a radial bearing and inner peripheral surface 30 of the sleeve. Further, with the rotation of this rotational member 28, the air which has flowed into the radial bearing flows into a pressure chamber 48 formed between the thrust end surface 34 of the fixed shaft 14 and the thrust end surface 38 of the thrust bearing member 26 cooperating therewith. A thrust gas bearing film is formed by the air flowing into the pressure chamber 48 and thus, the thrust end surface 38 of the thrust bearing member 26 is supported by this bearing film. If the rotational member 28 floats up, the air in the pressure chamber 48 flows outwardly through the axial hole 36.

As described above, in the present dynamic pressure gas bearing device for rotational unit, the rotational member 28 in rotation is supported in non-contact in the radial direction by a radial gas (air) bearing portion comprising a dynamic pressure groove bearing formed by the outer peripheral surface 16 of the fixed shaft and the inner peripheral surface 30 of the sleeve cooperating therewith and in the axial direction (thrust direction) by a thrust gas (air) bearing portion formed by the gas film created by the action of the dynamic pressure groove 18 of the radial gas bearing portion. Accordingly, the sleeve in rotation is kept in non-contact by the dynamic pressure gas bearing film and therefore, any irregularity of rotation caused by the bearing can be avoided.

Also, by the groove bearing being used as the radial bearing, a pre-pressure effect by the dynamic pressure acts in the radial direction and even during high-speed rotation, the radial vibration of the sleeve can be minimized. On the other hand, the thrust bearing portion creates a thrust load capability by utilizing the gas flowing out of the radial bearing portion due to the dynamic pressure effect to squeeze the gas flowing out through the axial hole 36 which opens substantially centrally of the thrust end surface 38, and this leads to a very simple construction and reduced cost.

Further, since one of the pair of thrust end surfaces 34 and 38 is formed into a convex spherical surface, the great starting torque which is a drawback of the dynamic pressure gas bearing can be reduced remarkably and the wear caused by the contact between the thrust end surface 34 and 38 during non-rotation can be reduced. Moreover, even in case wear powder is created by the thrust end surface contacting each other when the rotation is stopped, the wear powder is discharged outwardly of the bearing via the axial hole 36, and this leads to the advantage that the wear powder is prevented from accelerating the wear of the end surfaces.

Further, since the radial bearing portion and the thrust bearing portion are formed integrally with each other by the rotational member 28 and this rotational member 28 is supported by the cantilevered fixed shaft 14, the entire structure becomes simple and is hardly affected by the assembly accuracy and thus, there can be by provided a bearing device which is advantageous in both cost and accuracy. Also, the rotor 42 and stator 22 forming the drive motor are liable to heat, but since there is formed a gas stream which flows in through the bearing gap of the radial bearing portion due to the action of the dynamic pressure and flows out through the axial hole 36 opening into the thrust bearing portion, there is the advantage that a gas stream for cooling the motor can be autmatically formed.

The dynamic pressure generating groove provided in the radial bearing portion is designed in accordance with the conditions of use and therefore, the present invention may be carried out by using any other groove pattern than that shown in the above-described embodiment.

Second to fourth embodiments of the present invention will now be described. For simplicity, in these embodiments, the parts similar to those of the first embodiment are given similar reference numerals and need not be described but description will be made chiefly of different parts.

Referring to FIG. 2, a fixed shaft 14 has its upper end secured to a base plate 10 and is downwardly provided upright, and the fixed shaft 14 is clad with a rotational member 28 comprising a sleeve 24 and a thrust bearing member 26 pressure-fitted into the lower end thereof. The lower surface of the fixed shaft 14 is formed into a flat thrust end surface 34 while the upper surface of the thrust bearing member 26 is formed into a convex spherical thrust end surface 38. Opposed magnetic members 50 and 52 are fixed to the base plate 10 and the sleeve 24, and they constitute an auxiliary thrust magnetic bearing 54.

During non-rotation of the rotational member 28, the magnetic members 50 and 52 are attracted to each other and the rotational member 28 is not seated on but spaced apart from a cover 46, and the thrust end surfaces 34 and 38 are in contact with each other. When the rotational member 28 is rotated by a drive motor, the gas flowing in the direction of arrow A flows downwardly through a gap 32 between the outer peripheral surface 16 of the fixed shaft and the inner peripheral surface 30 of the sleeve and into a pressure chamber 48 and slightly forces the rotational member 28 downward by the pressure thereof. A thrust gas bearing film is then formed by the gas in the pressure chamber 48, and the trust end surface 38 is supported by this bearing film. According to the present embodiment, the rotational member 28 is upwardly biased by a force greater than weight thereof due to the action of the auxiliary thrust

bearing 54 and moreover, the difference between the two is smaller than the weight of the rotational member 28. Accordingly, as compared with the embodiment shown in FIG. 1, the wear of the thrust end surfaces 34 and 38 can be reduced during the stoppage of the rotational member 28.

The magnetic members 50 and 52 forming the auxiliary thrust bearing 54 may be either electromagnets or permanent magnets, but if a permanent magnet is used as at least one of these magnetic members, the structure thereof will become simple. Of course, permanent magnets may be used as both of these magnetic members or a combination of a permanent magnet and a magnetic member may be used.

The embodiment shown in FIG. 3 is essentially identical in construction to the first embodiment (see FIG. 1) except that a contrivance is made in the thrust bearing member 26 of the rotational member 28 and the portion above it. That is, a recess 60 is formed centrally of a relatively thick cover 46, a resilient member 62 such as a plate spring is mounted so as to cover the recess 60, and a thrust bearing member 66 having its lower surface formed into a flat thrust end surface 64 is fixed to the central portion of the resilient member 62. The diameter of the upper half of the axial hole 36 of the thrust bearing member 26 is enlarged and also, the upper surface thereof is formed into a thrust end surface comprising a convex spherical surface. The thrust bearing member 66 may be threadedly engaged with a bolt mounted to the cover 46 so that it is vertically movable.

The peculiar operation and effect of the present embodiment will now be described. During the repose and lowspeed rotation of the rotational member 28, the thrust bearing member 66 is urged against the thrust bearing member 26 by the action of the resilient member 62. Therefore, during the transportion of the bearing device, the rotational member is prevented from moving in the thrust direction to create backlash and damaging of the bearing device thereof can be prevented, and further during the low speed rotation pressure release through the bore 36 can be prevented.

Also, during rotation of the rotational member 28, the rotational member 28 floats up due to the pressure in the pressure chamber 48 and the thrust end surface 38 is supported by the thrust bearing film while, at the same time, the thrust bearing member 66 is raised upwardly by the pressure in the axial hole 36 against the action of the resilient member 62 and the thrust end surface 64 is also supported by the thrust bearing film. In this manner, the thrust bearing member 26, namely, the rotational member 28, is supported in its upper and lower surfaces by the bearing film and therefore, even in case extraneous vibration acts on the rotational unit, the vibration of the rotational member 28 provided with a polygon mirror 40 in the axial direction (thrust direction) can be minimized.

A dynamic pressure generating groove which will prevent outflow of the operating fluid may be formed in one of the thrust end surface 64 and the thrust end surface 68.

The embodiment shown in FIG. 4 is entirely similar to the second embodiment (see FIG. 21) except that a contrivance is made in the lower portion of the thrust bearing member 26 of the rotational member 28. The contrivance is that a cylindrical member 72 is partly embedded in the base plate 10 in opposed relationship with the thrust bearing member 26, and a coil spring 74 and a steel ball 76 are mounted in the hollow portion of the cylindrical member 72. The coil spring 74 and steel ball 76 are functionally similar to the resilient body 62 and thrust bearing member 66 in the third embodiment (see FIG. 3).

That is, during the repose and low-speed rotation of the rotational member 28, the steel ball 76 pushes up the rotational member 28 with the aid of the biasing force of the coil spring 74 and brings the thrust end surfaces 34 and 38 into contact with each other, while during high-speed rotation of the rotational member 28, the rotational member 28 is forced downward by the pressure of the air in the pressure chamber 48 and the thrust end surface 38 is supported by the thrust bearing film and also, the steel ball 76 is forced upward by the pressure in the axial hole 36 against the action of the spring 74 and thus, the thrust end surface 78 is supported by the thrust bearing film.

We claim:

1. A dynamic pressure assembly comprising as members within a closed housing:

a body;

a solid fixed shaft without a bore therethrough fixed to said body and having a free end;

a rotational member including a sleeve portion surrounding said fixed shaft with a radial clearance and a bottom portion having an internal surface opposed to an end surface of the free end of the fixed shaft, the rotational member being shiftable relative to the fixed shaft;

a pressure chamber formed between the internal surface of said bottom portion and the end surface of said free end;

means for rotating said rotational member, said rotating means including a rotor fixed to said rotational member at an axial position thereof and a stator opposed to the rotor and fixed to said body;

means for generating dynamic pressure in said radial clearance and supplying it to said pressure chamber through said radial clearance, the dynamic pressure generating means being formed between the shaft and the sleeve portion of the rotational member; and

pressure regulating means including a restricted flow passage which has an opening in the center of said internal surface to vent a pressurized flow from the pressure chamber, thereby regulating dynamic pressure therein, the rim of said opening defining an annular seat for engagement by a portion of the end surface for closing the opening;

the arrangement being such that said portion of the end surface of said shaft contacts said annular seat of the rotational member and closes the opening and thereby closes the passage when there is no relative rotation between the shaft and the sleeve or when such relative rotation is only at a low speed insufficient to develop in said pressure chamber a sufficiently high hydrodynamic pressure to axially displace the shaft relative to said housing by the hydrodynamic pressure established in said pressure chamber and opens the opening to said pressure chamber and thereby opens the passage when the rotation of the rotational member is at a high speed sufficient to develop a sufficiently high hydrodynamic pressure in said pressure chamber.

2. A dynamic pressure assembly according to claim 1, wherein said dynamic pressure generating means includes grooves formed in an outer peripheral surface of the fixed shaft.

3. A dynamic pressure assembly according to claim 1, further comprising a load member fixed to said rotational member at a position axially distant from said rotor.

4. A dynamic pressure assembly according to claim 3 wherein said load member is a polygon mirror and said housing is provided with a transparent portion through which light rays pass to reach the polygon mirror.

5. A dynamic pressure assembly as in claim 1 and further including preventing means for preventing axial movement of the rotational member when rotation of the rotational member is below a predetermined level.

6. A dynamic pressure assembly according to claim 5, wherein said preventing means allows rotation of the rotational member while preventing the axial movement thereof.

7. A dynamic pressure assembly according to claim 5, wherein said preventing means includes means for magnetically attracting the rotational member toward the housing to fix the rotational member axially.

8. A dynamic pressure assembly according to claim 5, wherein said preventing means includes biasing means for axially fixing the rotational member.

9. A dynamic pressure assembly according to claim 5, wherein said preventing means includes a spring and a ball disposed between the housing and the rotational member.

10. A dynamic pressure assembly according to claim 1, further comprising, a holding means for holding the rotational member relative to said body when no dynamic pressure is generated.

11. A dynamic pressure assembly according to claim 10, wherein said holding means comprises a pair of magnetic members disposed on a portion of the body and opposed portion of the rotational member.

12. A dynamic pressure assembly according to claim 11, wherein said holding means includes a deformable diaphragm fixed to the body to be touched to the rotational member.

13. A dynamic pressure assembly according to claim 11, wherein said holding means includes a ball to be touched to the rotational member and a spring to urge the ball toward the rotational member.

14. A dynamic pressure assembly comprising:

a body;

a solid fixed shaft without a bore therethrough fixed to said body and having a free end;

a rotational member including a sleeve portion surrounding said fixed shaft with a radial clearance and a bottom portion having an internal surface opposed to an end surface of the free end of the fixed shaft, the rotational member being shiftable relative to the fixed shaft;

a pressure chamber being formed between the internal surface of said bottom portion and the end surface of said free end;

means for rotating said rotational member, said rotating means including a rotor fixed to said rotational member at an axial position thereof and a stator opposed to the rotor and fixed to said body;

means for generating dynamic pressure in said radial clearance and supplying it to said pressure chamber through said radial clearance, the dynamic pressure generating means being formed between the shaft and the sleeve portion of the rotational member; and

pressure regulating means including a restricted flow passage which has an opening in the center of said internal surface to vent a pressurized flow from the pressure chamber, thereby regulating dynamic pressure therein, the rim of said opening defining an annular seat for engagement by a portion of the end surface for closing the opening;

the arrangement being such that said portion of the end surface of said shaft contacts said annular seat of the rotational member and closes the opening and thereby closes the passage when there is no relative rotation between the shaft and the sleeve or when such relative rotation is only at a low speed insufficient to develop in said pressure chamber a sufficiently high hydrodynamic pressure to axially displace the shaft relative to said sleeve by the hydrodynamic pressure established in said pressure chamber and opens the opening to said pressure chamber and thereby opens the passage when the rotation of the rotational member is at a high speed sufficient to develop a sufficiently high hydrodynamic pressure in said pressure chamber.

15. A dynamic pressure assembly according to claim 14, further comprising a load member fixed to said rotational member at a position axially distant from a position where said rotor is fixed.

16. A dynamic pressure assembly according to claim 15, wherein said load member includes a polygon mirror.

17. A dynamic pressure assembly comprising as members within a closed housing:

a body;

a solid fixed shaft without a bore therethrough fixed to said body and having a free end;

a rotational member including a sleeve portion surrounding said fixed shaft with a radial clearance and a bottom portion having an internal surface opposed to an end surface of the free end of the fixed shaft, the rotational member being shiftable relative to the fixed shaft;

a pressure chamber formed between the internal surface of said bottom portion and the end surface of said free end;

means for rotating said rotational member, said rotating means including a rotor fixed to said rotational member at an axial position thereof and a stator opposed to the rotor and fixed to said body;

a load member secured to said rotational member at a second axial position axially distant from said first position;

means for generating dynamic pressure in said radial clearance and supplying it to said pressure chamber through said radial clearance, the dynamic pressure generating means being formed between the shaft and the sleeve portion of the rotational member; and

pressure regulating means including a restricted flow passage which has an opening in the center of said internal surface to vent a pressurized flow from the pressure chamber, therby regulating dynamic pressure therein, the rim of said opening defining an annular seat for engagement by a portion of the end surface for closing the opening;

the arrangement being such that said portion of the end surface of said shaft contacts said annular seat of the rotational member and closes the opening and thereby closes the passage when there is no relative rotation between the shaft and the sleeve or when such relative rotation is only at a low speed insufficient to develop in said pressure cham-

ber a sufficiently high hydrodynamic pressure to axially displace the shaft relative to said sleeve by the hydrodynamic pressure established in said pressure chamber and opens the opening to said pressure chamber and thereby opens the passage when the rotation of the rotational member is at a high speed sufficient to develop a sufficiently high hydrodynamic pressure in said pressure chamber.

18. A dynamic pressure assembly according to claim 17, wherein said load member includes a polygon mirror.

19. A dynamic pressure assembly according to claim 18, wherein said closed housing is provided with a transparent member through which light rays from the exterior are transmitted to reach said polygon mirror.

20. A dynamic pressure assembly according to claim 19, wherein said transparent member is made of a glass.

* * * * *

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United States Patent [19]

Blaine

[11] **Patent Number:** **5,503,478**

[45] **Date of Patent:** **Apr. 2, 1996**

[54] **LUBRICANT DISTRIBUTION SYSTEM FOR BEARINGS AND BUSHINGS**

[75] Inventor: **Brad L. Blaine**, Grosse Pointe Woods, Mich.

[73] Assignee: **Federal-Mogul Corporation**, Southfield, Mich.

[21] Appl. No.: **415,277**

[22] Filed: **Apr. 3, 1995**

[51] **Int. Cl.**^{6} **F16C 32/06**

[52] **U.S. Cl.** **384/100**; 384/118; 384/286

[58] **Field of Search** 384/100, 114, 384/118, 119, 120, 286, 291

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Primary Examiner—Thomas R. Hannon
Attorney, Agent, or Firm—Lawrence J. Shurupoff

[57] **ABSTRACT**

A shaft bearing or bushing includes a hydrodynamic wedge film lubricant pumping surface along one edge region and an essentially continuous cylindrical land surface along the other opposed edge region. Lubricant moves circumferentially along the pumping surface and axially along the continuous land surface. The bearing provides extensive surface area oil film support for the rotating shaft and may be optimized for supporting a specific shaft load distribution.

8 Claims, 3 Drawing Sheets

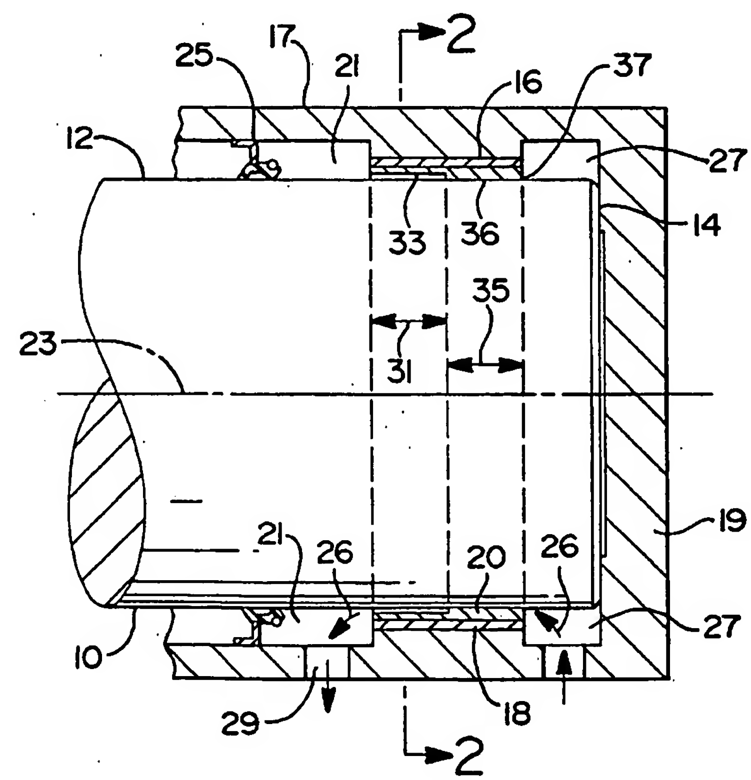
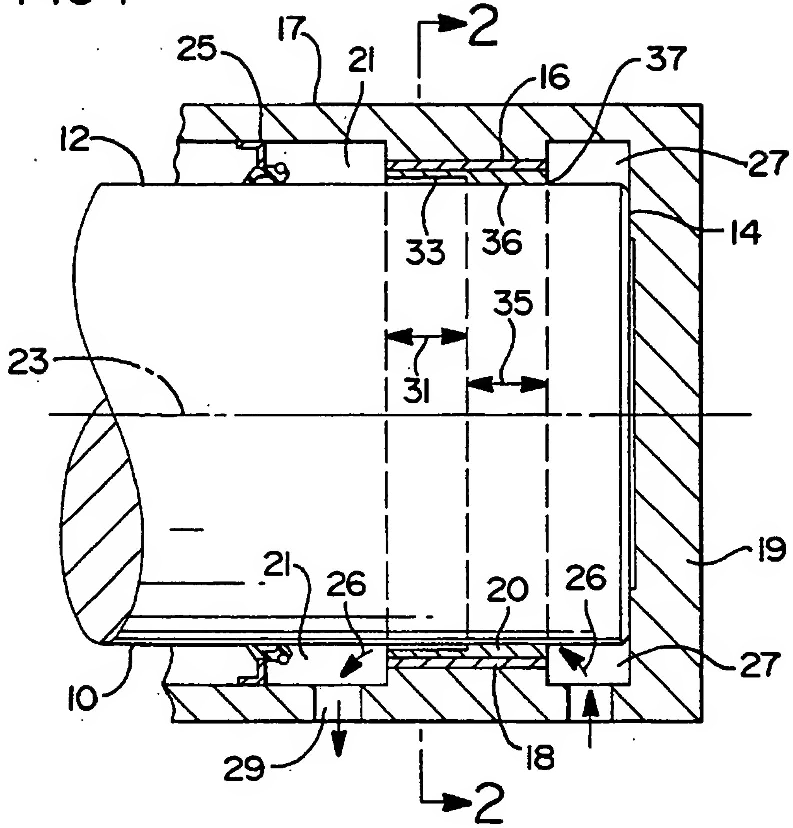


FIG 1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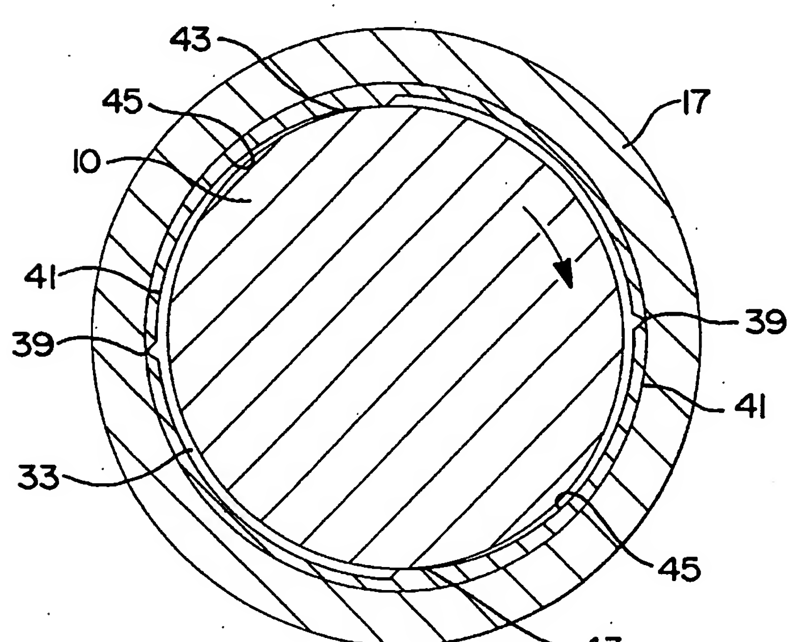
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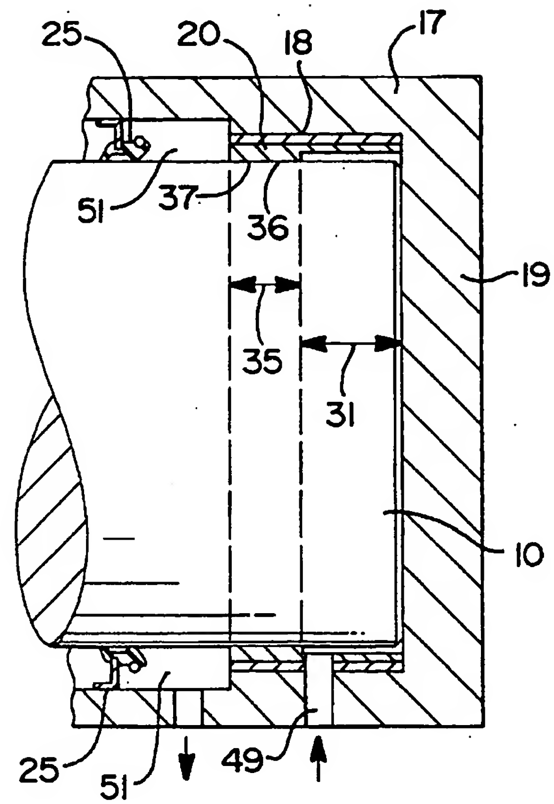
FIG 2

FIG 3



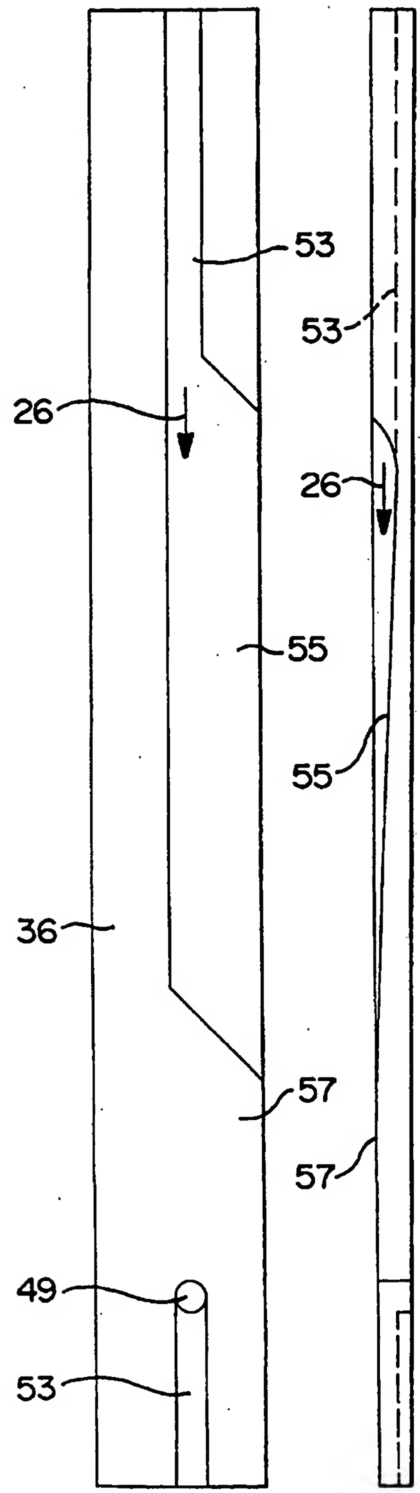
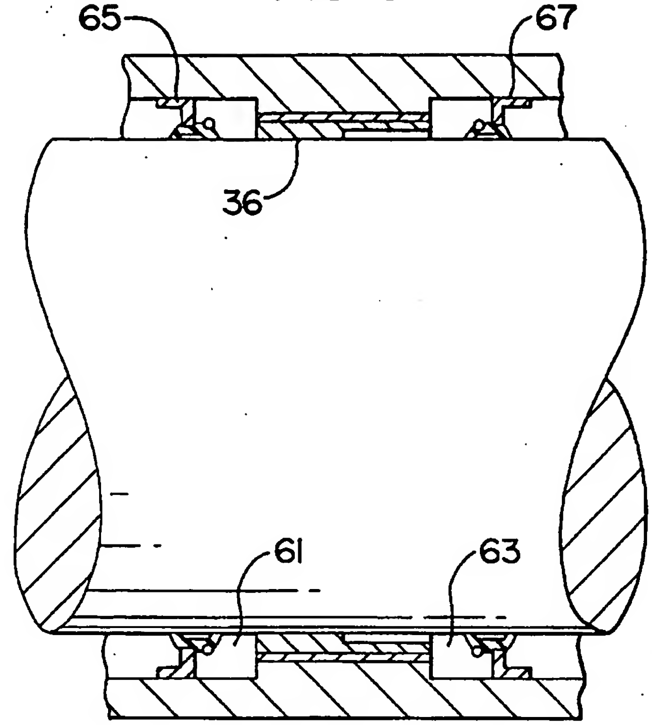


FIG 4 FIG 5

FIG 6



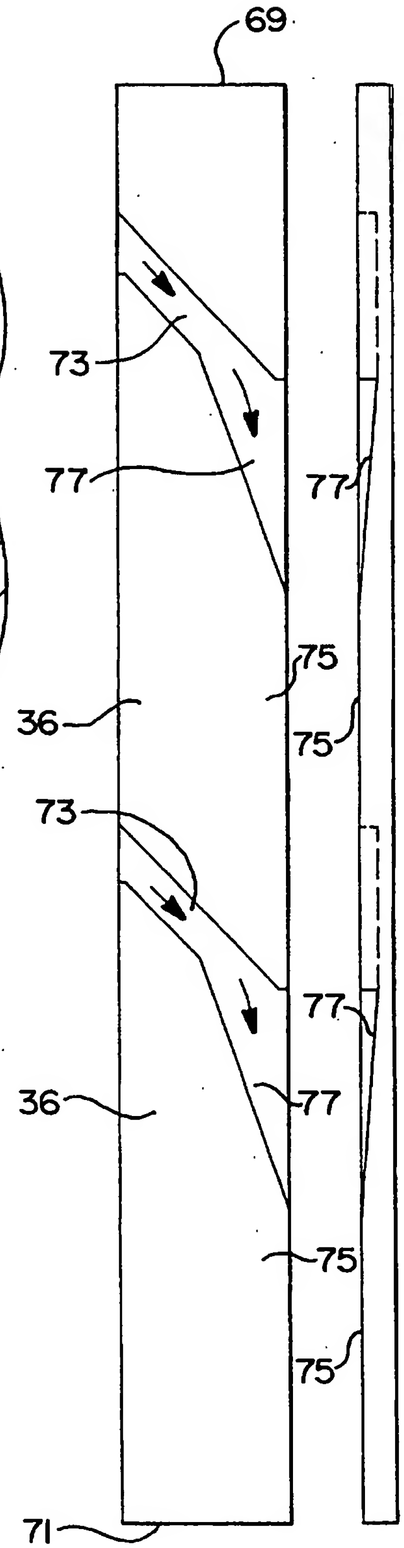


FIG 7 FIG 8

LUBRICANT DISTRIBUTION SYSTEM FOR BEARINGS AND BUSHINGS

BACKGROUND OF THE INVENTION

1. Field of the Invention

The present invention relates to a plain bearing or bushing having both a ramped and a cylindrical surface profile for distributing lubricating fluid to load-bearing surfaces under both hydrodynamic wedge film lubrication and conventional cylindrical film lubrication.

2. Description of Prior Developments

Rotational motion of a shaft is commonly used to maintain a load supporting lubricant film between annular support surfaces provided on the inner surfaces of bearings and bushings, (hereinafter collectively referred to as bushings) and the outer surface of the shaft. Rotation of the cylindrical shaft draws lubricant into one or more small clearance spaces between the shaft and the load bearing surfaces of the bushing.

The internal support surfaces of the bushing may include a fluid entrance surface spaced radially from the shaft side surface, a land surface having a minimal clearance with respect to the shaft surface and a concave ramp surface joining the fluid entrance surface and the land surface. Lubricating oil is drawn in a circumferential direction from the fluid entrance surface along the ramp surface and onto the land surface.

The oil is wedged into a small radial clearance between the land surface and the shaft side surface to form a pressurized load supporting film. The pressurized wedge-shaped oil film is not readily displaced out of the small clearance space, and is thus able to absorb or sustain relatively large radial shaft loads. This type of bearing lubrication is sometimes referred to as hydrodynamic wedge film lubrication.

Although hydrodynamic wedge film lubrication provides satisfactory results when used in many conventional bearing and bushing applications, the presence of ramped bearing surfaces over the full axial extent of the bearing or bushing decreases the stability provided to a rotating shaft as compared to a nonramped or nominally cylindrical bearing support surface. Moreover, such a ramped design produces a larger leak path for lubricant to escape from between the bearing and shaft.

U.S. Pat. No. 1,236,511 shows a bearing construction that utilizes wedge film lubrication of the above-mentioned type wherein oil is introduced to the bearing through four axially extending grooves in the bearing inner surface. As the shaft rotates, the oil clinging to the shaft surface is drawn circumferentially into small clearance spaces located midway between the grooves. The oil film established in the small clearance spaces provides a low friction support for the shaft, thereby protecting the shaft and bearing against contact and wear.

The circumferentially moving oil film is confined to circumferential motion by rims or lands that form shoulders along edge areas of the wedge film surfaces. A disadvantage of such rims is that if they are fully effective they can be in direct contact with the shaft surface, thereby producing frictional wear. Also, by confining the oil to a circumferential motion, any circulation of oil through the bearing is prevented because there is no convenient oil path out of the bearing. The oil will endlessly circulate in a circumferential direction around the bearing, thereby eventually heating the oil and generating carbon particulates.

Other patents showing wedge film lubrication achieved by circumferential oil motion are U.S. Pat. Nos. 2,631,905 and 3,680,932. In these patented arrangements, the edge areas of the wedge film surfaces are bounded by endless circular rim or land surfaces designed to prevent axial leakage of the circumferentially moving oil film away from the wedge film surfaces.

SUMMARY OF THE INVENTION

The present invention is directed to a radial bearing using wedge film lubrication together with a controlled axial flow of the wedge film oil through a cylindrical gap so that the oil circulates through the bearing in an expeditious manner.

The invention is primarily embodied in a sleeve bearing wherein approximately one-quarter to one-half the axial length of the bearing is internally ramped and contoured to form one or more wedge film pump surfaces designed to promote a circumferential flow of lubricant around the internal surface of the bearing. The remaining portion of the axial length of the bearing includes a cylindrical surface having a substantially uniform radial clearance relative to the shaft surface.

The lubricant moves circumferentially around the bearing while in contact with the wedge film surfaces and axially and/or helically while in contact with the cylindrical bearing surface. The cylindrical bearing surface establishes an essentially continuous load-supporting film around the shaft surface, thereby providing an essentially continuous uniform support action around the entire shaft circumference so as to stabilize the shaft. With the described arrangement, the unit loadings on the bearing are lessened as compared to a purely wedge-film bearing because the entire circumference of the bearing provides support to the shaft over a significant axial extent of the bearing.

The clearance between the cylindrical shaft surface and cylindrical bearing surface can be closely controlled by conventional machining or forming procedures so that the quantity of oil circulated through the bearing can be limited to a reasonable value consistent with the aim of controlling the thermodynamic effect.

By providing a hydrodynamic wedge effect over only a portion of the axial extent of the bushing or bearing, the bushing or bearing may be optimized to match its load handling capabilities with a particular shaft loading distribution while reducing lubricant leakage and improving shaft stability. That is, hydrodynamic ramps may be limited to that axial portion of the bearing which receives the greatest radial loads while the remaining axial extent of the bearing surface is of conventional circular or cylindrical shape for improving shaft stability and controlling or reducing axial lubricant flow and leakage.

Moreover, by providing a portion of the bearing with a full round cylindrical support surface, shaft support is increased so as to reduce stress concentrations and the breakdown of the lubricant film in the hydrodynamic wedge region, This in turn prevents metal-to-metal contact since the lubricant wedge is maintained.

Additional features and advantages of the invention will be further apparent from the attached drawings and drawing descriptions.

BRIEF DESCRIPTION OF THE DRAWINGS

FIG. 1 is a fragmentary elevational view of a rotary shaft supported by a radial bearing or bushing constructed according to the invention.

FIG. 2 is a transverse sectional view taken on line 2—2 in FIG. 1.

FIG. 3 is a fragmentary view taken in the same direction as FIG. 1, showing another embodiment of the invention.

FIG. 4 is a plan view of FIG. 3, unfolded to a flat condition to illustrate the contour of the bearing surface.

FIG. 5 is an edge view of the unfolded bearing of FIG. 4.

FIG. 6 is a view taken in the same direction as FIG. 1, illustrating a further embodiment of the invention.

FIGS. 7 and 8 are views taken in the same directions as FIGS. 4 and 5, illustrating the FIG. 6 bearing.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENTS

Referring to FIGS. 1 and 2, there is shown a shaft and bearing assembly constructed accordance with the present invention. A cylindrical or circular shaft **10** has a cylindrical surface **12** and a flat end surface **14**. As shown in FIG. 1., the shaft is supported in the radial direction by a sleeve-type bearing or bushing **16** which is press fit in housing **17**. End wall **19** of the housing is facially engaged with end surface **14** of the shaft to form an annular lubricant chamber **27** surrounding the shaft near its right end portion. The shaft is designed to rotate around the shaft axis **23**.

Bushing **16** includes an outer annular steel backing layer or strip **18** laminated to an annular layer or inner strip **20** of a bearing alloy having anti-friction properties, e.g., an aluminum-tin, an aluminum-lead or other suitable bearing alloy. A flat bimetal strip is typically curled into a circular configuration to form an annular bushing.

A conventional annular radial lip seal **25** is press fit in housing **17** to form an annular lubricant chamber **21** bordering the left end edge portion of bushing **16**. As indicated by the directional arrows, lubricating oil **26** is pumped into chamber **27** thereby pressurizing chamber **27**. The pressurized lubricant has a circumferential and axial motion as it passes leftwardly through the clearance space between bushing **16** into the annular chamber **21**. The oil **26** passes out of chamber **21** through an exit opening **29**.

The left axial half portion of bushing **16** designated by numeral **31** is internally contoured to form a wedge film pump surface designated generally by numeral **33** in FIG. 2. This left portion of the bushing experiences the greatest deflection and loading. The right half portion of bushing **16** designated by numeral **35** in FIG. 1 has a cylindrical internal surface **36** having a uniform radial clearance relative to the shaft side surface **12**. This right portion of the bushing experiences a lower load distribution than the left portion.

In FIG. 1, the uniform radial clearance is designated by numeral **37**. Typically, clearance **37** is about 0.001 inch, whereby a relatively thin load supporting oil film is maintained in the clearance space while the oil is moving in a right-to-left direction as shown in FIG. 1.

Wedge film pump surface **33** includes two axial grooves **39** for supplying lubricating fluid to two fluid entrance surfaces **41** spaced radially outwardly from the cylindrical shaft surface **12**. Pump surface **33** further includes two cylindrical portion land surfaces **43** that are coextensive or coplanar with cylindrical surface **36** constituting the right half of the bushing.

Land surfaces **43** promote shaft stability while maintaining the lubricant in a highly pressurized condition. An arcuate ramp surface **45** joins fluid entrance surface **41** to land surface **43**. Although land surfaces **43** can be quite narrow and essentially define a line contact with the shaft, it is preferred to extend land surfaces **43** circumferentially over at least several degrees of arc, i.e. 2 to 5 degrees, so as to provide a portion of a cylindrical support surface which reduces shaft whip and promotes shaft stability.

As shaft **10** rotates in a clockwise direction as shown in FIG. 2, the circumferentially moving shaft surface draws oil along the ramp surface **45** to form a wedge type oil film on land surface **43**. Simultaneously, the pressurized condition of lubricant chamber **27** tends to move the oil axially in a right-to-left direction across the cylindrical bushing surface **36** and onto the ramped region **31**.

The oil is introduced onto the ramped region **36** primarily at or near the groove **39** and is distributed through the ramp of bushing surface **41** by the movement of cylindrical shaft surface **12**. The oil is distributed uniformly on land surface **43** thereby increasing its load carrying capability. The oil on surfaces **36** and **43** provides the desired lubricant support for shaft **10**. The system is designed to provide a continuous circulation of the oil through the bearing or bushing while the shaft is rotating.

By aligning and mounting the ramped portion **31** of the bushing adjacent that section of the shaft which produces the greatest unit loading, such as by shaft whip or other deflection, the bushing may be optimized for maximum loading performance. Since the ramped portion **31** can withstand greater dynamic loading than cylindrical portion **35**, overall bearing performance may be increased. In addition, the cylindrical portion **35** provides greater stability than that normally possible with only the hydrodynamic portion **31**, and thereby reduces the peak loading applied over portion **35**.

The bushing performance can be even further optimized by circumferentially aligning the land surfaces **43** with that circumferential portion or portions of the shaft surface which experiences the greatest loading. For example, if it is known that the greatest radial loading on a particular shaft occurs at the 6 o'clock and 12 o'clock positions, the bushing would be aligned as shown in FIG. 2 such that land surfaces **43** are located adjacent the corresponding 6 o'clock and 12 o'clock circumferential locations.

FIGS. 3 through 5 illustrate a second shaft bearing arrangement that is generally similar in a functional sense to the system depicted in FIGS. 1 and 2. However, in the FIG. 3 system, the pressurized lubricant is introduced to the wedge film pump surface through a hole **49**. The oil has a right-to-left motion through the bushing in addition to the desired circumferential motion.

In FIG. 3, the right portion of the bushing designated by numeral **31** is internally contoured to form the wedge film pump surface. The left portion of the bushing designated by numeral **35** has a cylindrical internal surface **36** having a uniform radial clearance relative to the shaft side surface. Oil discharged from the clearance **37** flows into an annular chamber **51** defined by the left end edge of the bushing and an annular lip seal **25**. Seal **25** and end wall **19** form fluid barriers or confinement mechanisms for the oil circulated through the bushing.

As shown in FIGS. 4 and 5, the wedge film fluid pump surface portion **31** includes a fluid entrance surface **53** in the form of a groove or channel adapted to be spaced or recessed from the shaft surface, a ramp surface **55** extending from surface **53** and a land surface **57** axially and radially coextensive or coplanar with the bushing surface **36** defining the left portion of the bushing axial dimension. In the FIG. 3 arrangement, the wedge film pump surface is located within

or forms an annular lubricant inlet chamber surrounding the shaft **10**. Lubricant is introduced to the pump surface through hole **49**.

As the shaft rotates around the shaft axis, oil is drawn circumferentially along the ramp surface **55** onto land surface **57**. The pressurized condition of the rightmost chamber defined by the wedge film pump surface causes the oil to have a leftward motion across the bushing cylindrical surface **36**. The shaft loading is supported on the oil film established on surfaces **36** and **57**. The bearing system of FIG. **3** operates in approximately the same fashion as the system depicted in FIG. **1**.

FIGS. **6** through **8** show a shaft bearing system according to the invention designed to use a self-contained supply of lubricant, i.e. a bearing wherein there is no external lubricant source. In FIG. **6**, the lubricant has two-directional flow through the bushing as determined by the relative pressures in two annular chambers **61** and **63** bordering the left and right edge portions of the bushing. Chambers **61** and **63** are defined by two fluid-confinement lip seals **65** and **67**.

FIGS. **7** and **8** show the bushing unfolded to a flat condition. In the actual bushing, as depicted in FIG. **6**, the axial edges **69** and **71** of the bushing wall are abutted and secured together to form an endless tubular bushing sleeve. The bushing has wedge film pump surfaces that define fluid entrance surfaces **73** which are spaced or recessed from the shaft surface **12** as shown in FIG. **6**, land surfaces **75** extending axially and radially coplanar or coextensive with the inner bushing load support surface **36** and arcuate sloping ramp surfaces **77** joining surfaces **73** to land surfaces **75**.

FIG. **7** includes arrows denoting the flow of lubricant **26** along surfaces **73**, **77** and **75** during rotational motion of the shaft. The shaft circumferential motion draws the lubricant along ramp surfaces **77** onto land surfaces **75**. The flow of oil along surface is **73** and **77** transfers oil from chamber **61** to chamber **63**, thereby pressurizing chamber **63** while depressurizing chamber **61**. A pressure differential is thus established tending to cause oil on bearing surface **36** to move in a right-to-left direction as shown in FIG. **6**. An internal oil circuit is established across the bushing.

The bearing system of FIG. **7** operates in essentially the same fashion as the systems of FIGS. **1** and **3**, except that the lubricant supply is self-contained, i.e. not remote from the shaft support.

Bearing systems of the present invention can be used at one or both ends of a rotary shaft, as shown in FIGS. **1** and **3**. Also, such bearings can be used at intermediate points along a shaft, as shown in FIG. **6**. When the bearing is supported at an end of the rotary shaft, the bearing advantageously adopts to orbital shaft motions of the shaft produced by transverse loadings at intermediate points along the shaft. In this case, the cylindrical bearing surface **36** is located closest to the end surface **14** of the shaft.

The continuous circumferential shaft support provided by the oil films on the cylindrical bearing surfaces **36** absorbs the multi-directional load forces sometime is associated with orbital shaft motions, i.e. minor cyclic motions of the shaft surface toward or away from the shaft axis. Surfaces **35** and **36** may be of differing axial length to suit a particular shaft load distribution.

The drawings show specific forms of the invention. However, it will be understood that the invention can be practiced in various forms.

What is claimed is:

1. A shaft and bushing assembly, comprising:

a rotary shaft having a cylindrical side surface;

a bushing encircling said shaft, said bushing having first and second axial end portions;

first fluid confinement means cooperable with the first end portion of said bushing to form a first annular lubricant chamber surrounding the shaft;

second fluid confinement means cooperable with the second end portion of said bushing to form a second annular lubricant chamber surrounding said shaft;

said bushing having an internal surface facing the cylindrical side surface of the shaft; the internal surface of said bushing comprising a cylindrical surface contiguous with said second end portion of the bushing, said cylindrical surface having a uniform radial clearance relative to the shaft surface; and

the internal surface of said bushing further comprising a wedge film fluid pump surface contiguous with said first end portion of the bushing, whereby rotational motion of the shaft causes lubricant to be pumped from the first lubricant chamber across the fluid pump surface and onto the cylindrical surface of the bushing.

2. The shaft and bushing assembly of claim **1**, wherein said wedge film fluid pump surface comprises a fluid entrance surface spaced from said shaft surface, a land surface coplanar with said cylindrical surface, and a ramp surface joining said fluid entrance surface and said land surface whereby the rotation of the shaft causes lubricant to be drawn along the ramp surface onto the land surface.

3. The shaft and bushing assembly of claim **1**, wherein said bushing has an axial dimension defined as the axial distance between the bushing end portions; the cylindrical surface of said bushing having an axial dimension that is approximately three-quarters to one-half the axial dimension of the bushing.

4. The shaft and bushing assembly of claim **3**, wherein said wedge film fluid pump surface has an axial dimension that is approximately one quarter to one-half the axial dimension of the bushing.

5. The shaft and bushing assembly of claim **1**, wherein said shaft has an end surface normal to said cylindrical side surface; said bushing being spaced from the shaft end surface so that one of the fluid chambers is located between the shaft end surface and the bushing.

6. The shaft and bushing assembly of claim **1**, wherein one of said fluid confinement means comprises an annular seal encircling the rotary shaft.

7. The shaft and bushing assembly of claim **1**, and further comprising means for admitting pressurized lubricant to said first lubricant chamber; said admitting means comprising a radial hole extending through said bushing so as to communicate with said fluid entrance surface.

8. A shaft bushing comprising an annular steel backing layer laminated to an inner bearing alloy layer, said alloy layer comprising a contoured ramped surface portion extending axially from one axial end portion of said bushing toward an axial inner portion of said bushing and a first cylindrical surface portion extending from an opposed axial end portion of said bushing toward said axial inner portion of said bushing, said contoured ramped surface portion comprising a fluid entrance portion recessed with respect to said first cylindrical surface portion and extending circumferentially to a land surface, said land surface defining a second cylindrical surface portion which extends axially and radially coextensively with said first cylindrical surface portion.

* * * * *

United States Patent [19]

Adolfsson et al.

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[45] **Date of Patent: Oct. 1, 1991**

[54] **MAGNETIC BEARING BUSHING I**

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[58] Field of Search 384/133, 114, 115, 117, 384/118, 125, 280, 286, 291, 309, 311, 312, 312, 397, 368, 371, 192, 313, 316

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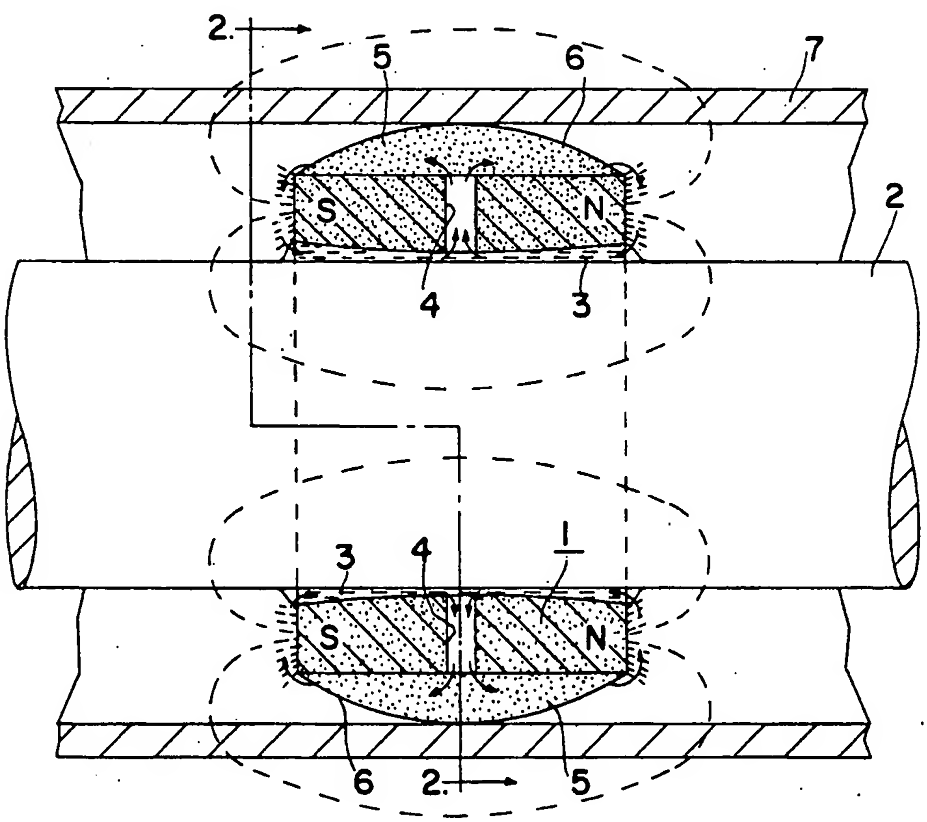
[57] **ABSTRACT**

A bearing bushing incorporating a sleeve (1) of magnetizable material forming the sliding surface of the bearing, a shaft (2) mounted in said sleeve (1) and a lubricant in the form of a magnetic fluid (3) between the sliding surface of the sleeve (1) and the shaft (2), whereby the sleeve (1) is magnetized in an axial direction. For causing a hydrodynamic lubrication and an increased cooling effect the sleeve (1) has a number of radial holes (4) through which the magnetic fluid (3) may pass to the other side of the sleeve (1), and which, by the axial magnetic field about the sleeve (1), is given a circulating motion along the inner and outer side of the sleeve.

4 Claims, 1 Drawing Sheet



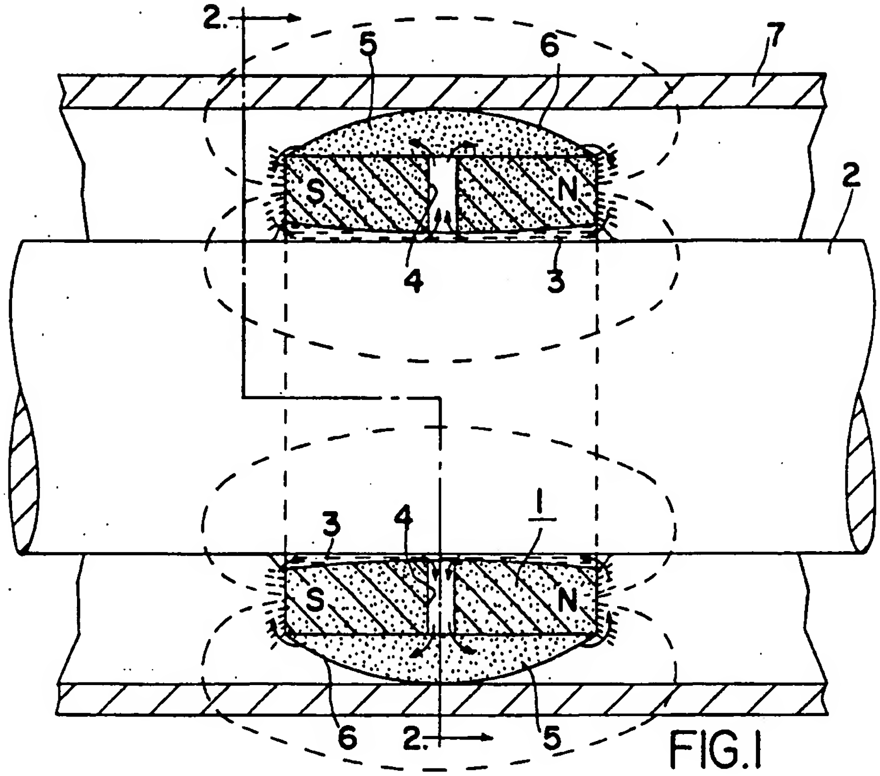


FIG.1

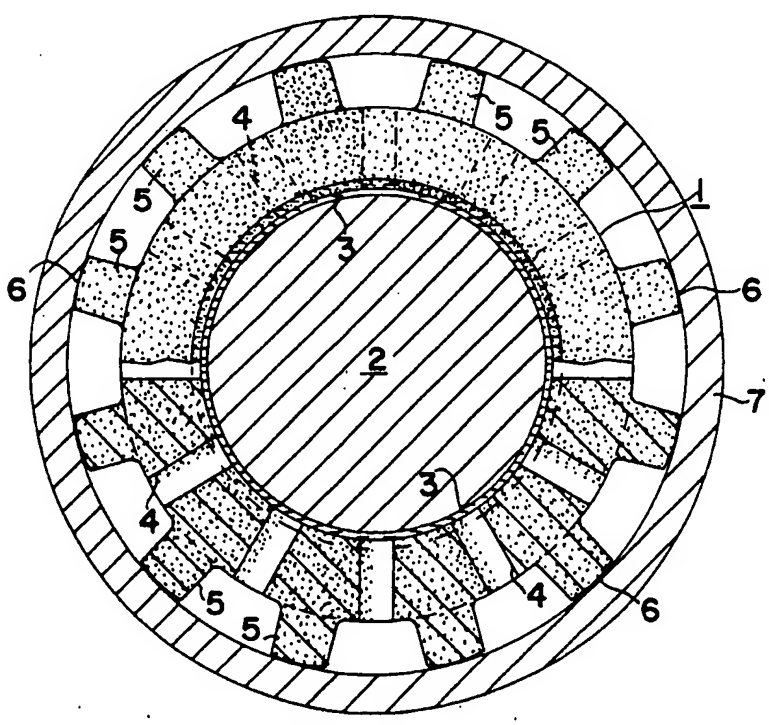


FIG.2

MAGNETIC BEARING BUSHING I

FIELD OF THE INVENTION

The present invention relates to a bearing bushing incorporating a sleeve of magnetizable material forming the sliding surface of the bearing, a shaft mounted in said sleeve and a lubricant in the form of a magnetic fluid between the sliding surface of the sleeve and the shaft, whereby the sleeve is magnetized in an axial direction.

BACKGROUND OF THE INVENTION

Magnetic bearings of the above mentioned type are used in many applications, e.g. at disk storages for computers and in domestic machines. Bearings of this type are used in these applications to achieve silent operation, extended operation life, and high rotational accuracy. An important advantage of such bearings is that lubricant leakage is prevented because the magnetic field generated by the magnetic bearing sleeve retains the lubricant, which contains mangetizable particle material.

In DE-A-3.304.623 is shown a magnetic bearing incorporating a magnetic bearing sleeve, which on its surface facing the shaft has grooves for causing a hydrodynamic pressure in the magnetic fluid. This device has a complex design, and the shape of the grooves furthermore requires that the shaft be continuously rotated in the same direction.

SUMMARY OF THE INVENTION

The purpose of the present invention is to provide a bearing of the above type characterized by novel features of construction and arrangement to provide hydrodynamic lubrication and efficient cooling, and which is simple and inexpensive to manufacture. This is achieved by providing a sleeve having a number of radial holes through which magnetic fluid may pass to the outer side of the sleeve and through an axial magnetic field around the sleeve. This arrangement produces a circulating motion of the magnetic fluid along the inner and outer side of the sleeve.

BRIEF DESCRIPTION OF THE DRAWINGS

These and other objects of the present invention and the various features and details of the operation and construction thereof are hereinafter more fully set forth with reference to the accompanying drawings, wherein:

FIG. 1 is an axial section through a bearing bushing according to the invention and having a shaft positioned therein.

FIG. 2 is a section along line 2—2 in FIG. 1.

DETAILED DESCRIPTION OF THE PREFERRED EMBODIMENT

The bearing bushing generally designated B incorporates a sleeve 1 of magnetizable material, preferably a synthetic material, in which have been embedded magnetic particles, e.g. magnetite particles. The sleeve 1 is mounted on a shaft 2. The shaft 2 may be of a magnetic or non-magnetic material. Between the sleeve 1 and the shaft 2 is introduced a lubricant in the form of a magnetic fluid 3, a so called "ferro-fluid", i.e. a colloidal dispersion or suspension of small magnetic particles in a carrier fluid, such as oil. The magnetic particles are retained in stable colloidal suspension by means of a dispersant. Such magnetic fluids can be introduced and retained in such spaces without use of a physical container by means of the magnetic field.

The sleeve 1 is magnetized in an axial direction, i.e. the magnetic fluid 3 gathers at the ends of the sleeve 1 thereby sealing it off. As shown in the drawing, the sleeve 1 has a number of radial holes 4 through which magnetic fluid 3 may pass to the outer side of the sleeve 1. The axial magnetic field around the sleeve 1 induces circulation of the magnetic fluid 3 which provides a cooling effect.

In order to further improve this cooling effect the sleeve 1 has external radial cooling flanges 5. The radial holes 4 are preferably located between the external cooling flanges 5. The external cooling flanges 5 preferably have spherical set up surfaces 6 for the sleeve 1 in a bearing housing 7, or the like. The sleeve 1 furthermore preferably has a cambered inner form for taking up shaft obliquities.

As mentioned above, the sleeve 1 is manufactured from a synthetic material, e.g. plastic or rubber with embedded magnetite particles or an embedded magnetizable steel spring. The advantages of a sleeve of magnetizable plastic material are obvious, e.g. it may be manufactured by injection molding and it is possible to give it complex shapes.

What is claimed is:

1. A bearing bushing incorporating a sleeve (1) of magnetized material forming the sliding surface of the bearing and having external axial cooling flanges (5), a shaft (2) mounted in said sleeve (1) and a lubricant in the form of a magnetic fluid (3) between the sliding surface of the sleeve (1) and the shaft (2), whereby the sleeve (1) is magnetized in an axial direction.

2. The bearing bushing as claimed in claim 1, characterized by radial holes (4) situated between the cooling flanges (5).

3. The bearing bushing as claimed in claim 1, characterized therein, that the cooling flanges (5) have spherical surface portions (6) forming a spherical set up surface for the sleeve (1).

4. A bearing bushing incorporating a sleeve (1) of magnetized material forming the sliding surface of the bearing, a shaft (2) mounted in said sleeve (1) and a lubricant in the form of a magnetic fluid (3) between the sliding surface of the sleeve (1) and the shaft (2), whereby the sleeve (1) is magnetized in axial direction, characterized therein, that the sleeve (1) has a number of radial holes (4) through which the magnetic fluid (3) may pass to the outer side of the sleeve (1), and which, by the axial magnetic field around the sleeve (1), is given a circulating motion along the inner and outer side of the sleeve.

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